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INVESTIGATION OF ADVANCED HELICOPTER STRUCTURAL DESIGNS

Volume II - Free Planetary Transmission Drive

Sikorsky Aircraft
Division of United Technologies Corporation
Stratford, Conn. 06602

May 1976

Final Report

Approved for public release;
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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This effort is one of two parallel contractual studies to define advanced structural configurations, advanced materials, and fabrication technology to satisfy requirements for a complete helicopter. The associated study program was conducted by Boeing-Vertol under the terms of Contract DAAJ02-74-C-0066.

Numerous design concepts, material selections, and manufacturing techniques were investigated for the various helicopter components (e.g., body group, main rotor, and transmission). The best overall concepts were selected and integrated into a complete advanced helicopter design, with predictions of improved weight, cost, and aircraft performance.

Mr. L. Thomas Mazza, Technology Application Division, served as project engineer, with Mr. E. Rouzee Givens directing the "Free Planetary Transmission Drive" study portion of the program.

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Volume I of this report is the application of advanced concepts and materials to a medium-sized utility transport helicopter. This volume investigates the free planetary transmission as applied to a utility transport helicopter.

High-power rotary speed reduction gearing in helicopter transmissions usually takes the form of conventional epicyclic planetary reduction units. Research and development has led to development of the free planet over

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transmission concept. This type of drive, compared with conventional two-stage planetary transmissions, promises several advantages, including 10% lighter weight, 16.5% lower cost in 500-unit quantities, 1/2 percent greater efficiency, almost twice the reliability, and greater tolerance to loss of lubrication.

The free planet transmission concept is defined as a planetary gear arrangement in which the planets are not constrained by being secured to a spider or planetary carrier. This lack of constraint is achieved by satisfying certain geometric relationships with conventional compound planetary gearing.

This study commenced with a survey of current work in free planet transmissions and a review of the actual hardware used in a 500-HP development test. This review indicated the attractiveness of the free planet transmission concept.

During preliminary design of a free planet transmission for the Medium Utility Transport (MUT) aircraft, design requirements were established for power, speeds, rotations, and size of envelope. Various drive train arrangements were considered, including high-speed bevel gear inputs, high-speed helical gear inputs, and dual high-speed spur gear inputs. The high-speed spur gear arrangement was selected as best for the MUT aircraft. A free planet transmission was also examined for a UTTAS-type drive arrangement.

Design analysis for the free planet was developed for gear tooth stress, axial length, pinion shaft design, and roller ring loads. This design analysis was the basis for development of a computer program for selecting the free planet design. The gear geometry parameters were varied, and some simple parametric curves were developed. Preliminary design layouts were made of both a conventional two-stage planetary and a free planetary transmission.

A final design layout was made of the free planet transmission, and cost, weight, survivability/vulnerability, and reliability characteristics were determined. Through use of a helicopter design model, the impact of the free planet transmission on aircraft performance was determined.

It is recommended that a free planet transmission be built for a helicopter application in which a single-engine 20,000 to 30,000 rpm input is available at a design power of 400 to 500 HP. This would permit verification of the concept and demonstration of its projected improvements.

PREFACE

The Program reported herein was conducted during a seven-month period from 20 January 1975 to 20 July 1975 for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USAAMRDL), Fort Eustis, Virginia, under Contract DAAJ02-74-C-0061.

USAAMRDL technical direction was provided by Mr. L. Thomas Mazza and Mr. E. Rouzee Givens of the Eustis Directorate, Technology Applications Division.

The program was conducted at Sikorsky Aircraft, Stratford, Connecticut, under the technical supervision of Mr. M. J. Rich, Sikorsky Aircraft, Structures and Materials Branch. Principal investigator was Mr. A. Korzun of the Transmission Design and Development Section.

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INTRODUCTION

Transmissions employing conventional planetary gear drives have performed satisfactorily for many years. Continuing research and development has been aimed at improving the performance of such transmissions. One promising development resulting from this effort has been the free planet transmission. This transmission can be classified as a quasi-compound planetary employing a sun gear, planet spindle assemblies, ring gears, and rolling rings.

The free planet concept was developed in the 1960's by The Curtiss-Wright Corporation. The concept covers broadly those planetary gear arrangements in which the planets are not constrained by being secured to a carrier. This lack of constraint was achieved by balancing moments and forces in the various planes through use of conventional compound planetary gearing.

Initial work was done with the Curtiss-Wright Power Hinge,[R] which has been used satisfactorily for fixed-wing aircraft flap actuation systems. Figure 1 illustrates the power hinge concept.

In 1970, Sikorsky Aircraft Division conducted an engineering design study to evaluate advances in VTOL aircraft drive train technology (Reference 1). Included in this evaluation was a free planet transmission for a 4,000-HP helicopter drive train. The configuration investigated offered potential advantages over a conventional planet by:

(1) eliminating planet bearing power losses and failures,

(2) having low planetary weight,

(3) permitting high reduction in two compound stages of high efficiency,

(4) providing sufficient flexibility and self-centering to give good load distribution between planet pinions,

(5) effectively isolating planetary elements from deflections of housing, and

(6) increasing operating time after loss of lubricant, since there were no planet bearings.

In 1972 and 1973, Curtiss-Wright designed, fabricated, and tested two 500-HP, 20-to-1 reduction ratio single-stage free planet transmissions (Reference 2). Testing was accomplished through a regenerative arrangement and indicated high mechanical efficiency, good load distribution, and potential advantages in weight reduction, reliability, survivability, and cost.

In January 1975, Sikorsky Aircraft and Boeing Vertol received contract modifications to a program for Advanced Helicopter Structural Design Investigation, to evaluate free planet transmission preliminary designs for helicopter application. This section of the report presents the results of the Sikorsky evaluation.

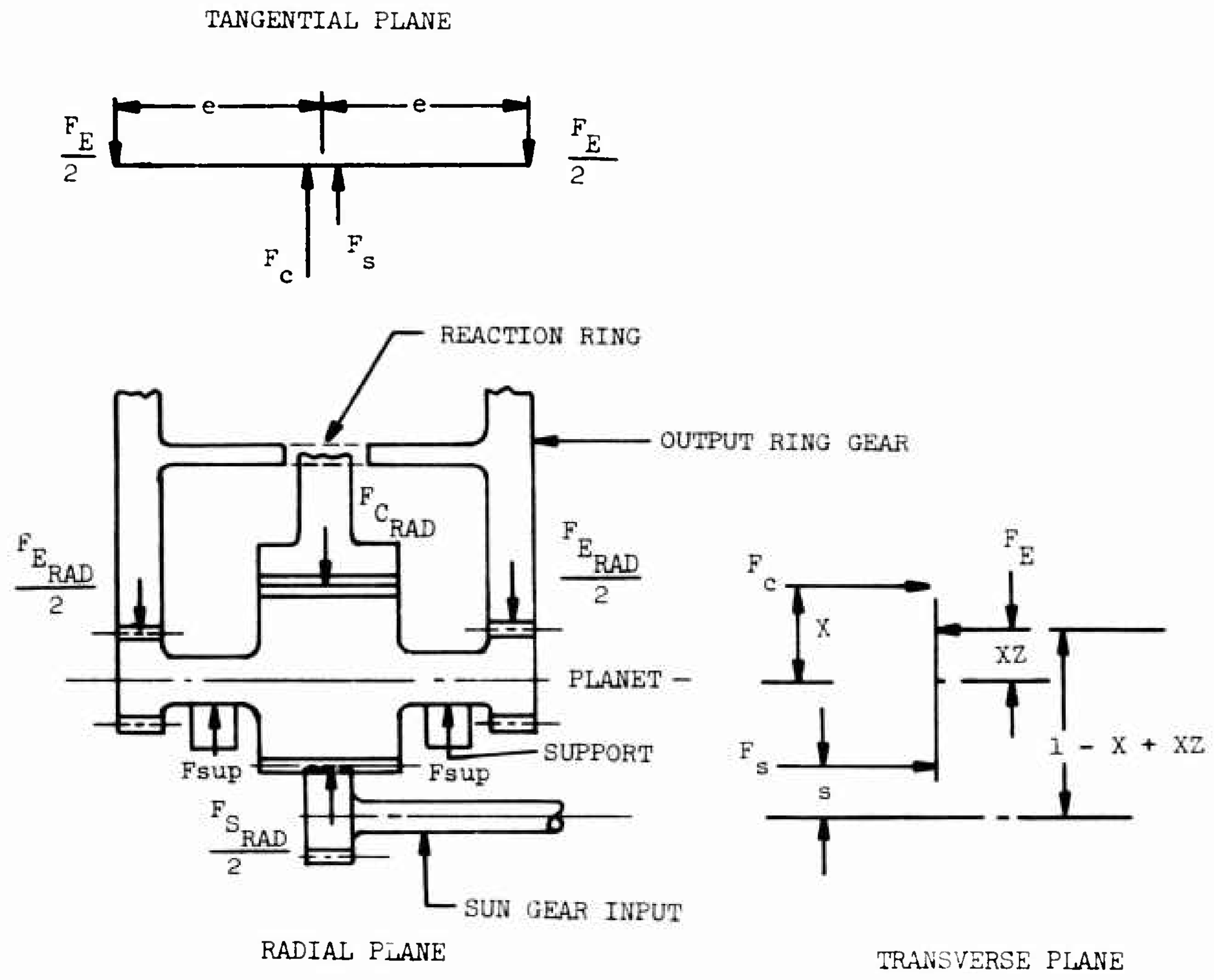


Figure 1. Power Hinge Concept

PRINCIPLE OF OPERATION OF FREE PLANET DRIVE

The free planet transmission concept covers a variety of planetary gear configurations employing no planet carriers or conventional planet mounting bearings. The free planet transmission is so configured that the forces on three orthogonal planes through the planet pinion shaft are in equilibrium.

The concept is best illustrated by first studying a simple conventional compound planetary drive (Figure 2). This design requires bearings to react loads in the tangential and radial planes. The forces in the transverse plane are already in equilibrium as a result of the reduction ratio. The bearing load in the tangential plane is approximately 5 to 6 times that in the radial plane.

The free planet design evolved from recognition of the advantages afforded through elimination of the bearings, which react loads in the tangential plane. The bearings could be eliminated by separating the gears in an axial direction.

Figure 3 is a schematic of a free planet drive that requires no conventional rolling element. The planet gear faces are spaced axially to enable the gear tooth forces to keep the planet spindles in equilibrium. The gears must be so spaced that they lie along the balance line. The gear tooth separating and centrifugal forces are reacted by and balanced out by cylindrical rings concentric with the sun gear axis. The planet spindles have diameters that roll freely on the cylindrical rings. The planet spindles are free in the sense that they are constrained only by the gear meshes and the free-floating cylindrical support rings.

One can verify that all forces and moments add up to zero about any point in or parallel to the three planes.

The free planet drive concept, as it has evolved, can be summarized as follows:

(1) The reduction ratio requirement and maximum diameter define the forces and the geometry in the transverse plane.

(2) Free-floating rings react the loads in the radial plane.

(3) Skewing moments in the tangential plane are eliminated by spacing the gears axially.

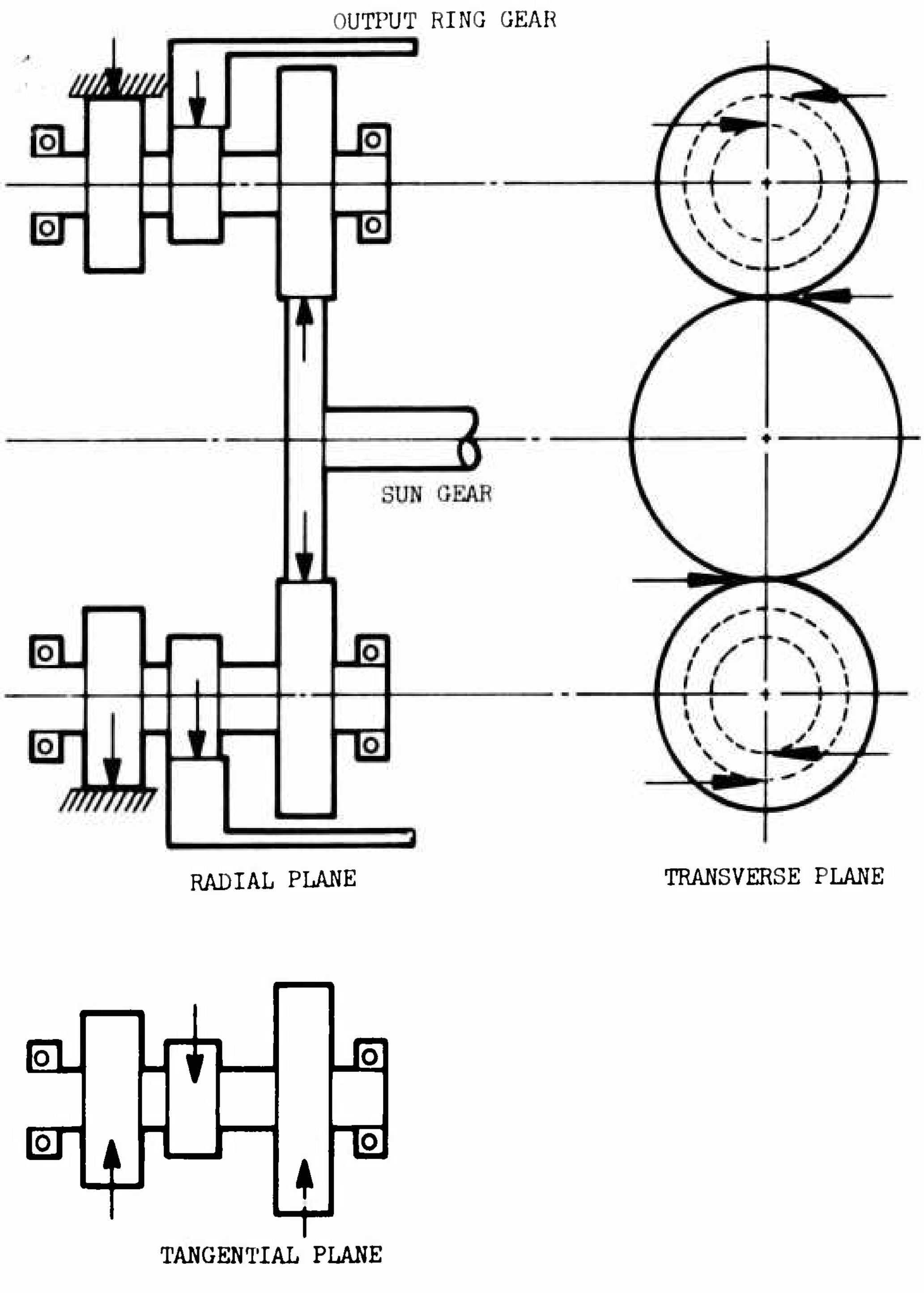


Figure 2. Conventional Compound Planetary Drive

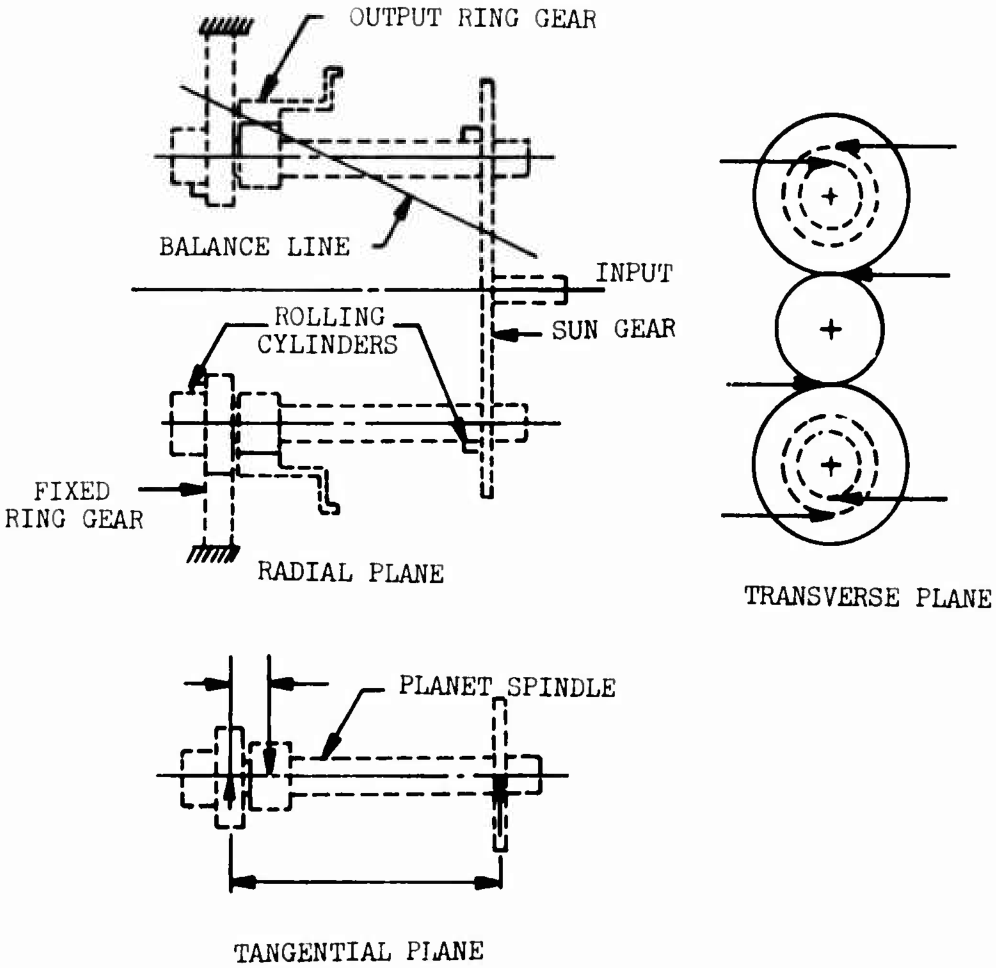


Figure 3. Schematic Free Planet Transmission

PREVIOUS TEST EXPERIENCE

Demonstration free planet test hardware designated FP500 and FP501 was designed, fabricated, and tested by Curtiss-Wright. The FP501 unit is shown in Figure 4. The unit is a compound planetary gear assembly consisting of three gear meshes. The first plane of the planet gear meshes with the sun gear. The second plane meshes with the output internal ring gear. The third plane meshes with the stationary internal ring gear. The first and second planes of the planet gears are splined, piloted, and locked to a quill shaft by a nut and cup lock. The third plane is splined, double piloted, and locked to the second plane of the planet gear by a nut and cup lock. The gears are timed so that the second and third planes are aligned. The FP501 unit is the same as the FP500 except that it has no quill, and torque is transmitted through the hollow support shaft.

Static Test Results

Static tests indicated good load distribution between planet spindles and gear tooth load patterns. The stiffer FP501 unit resulted in a wider spread of load distribution at lower loads. Gear meshing patterns indicated full face contact, which led to the conclusion that there was no end loading, thus verifying the predicted self-alignment under static conditions.

Dynamic Test Results

A 50-hour endurance test was run in a back-to-back, or regenerative, test facility at a rated speed of 8,000 rpm and power of 500 HP. During routine inspection after 26.75 hours, fretting and wear of the gear pilots were observed. Corrective action was taken to permit completion of the 50 hours of testing. Splines, gear, and shaft pilot were cleaned and plated to give a tighter fit. Final teardown after 50 hours of testing indicated that the free planet components were in excellent condition and there was no further deterioration of the splines and shaft pilots (Reference 2).

The FP501 unit (without quill shaft and with pinion torque transmitted through the hollow support shaft) was subjected to 9-3/4 hours of testing in a regenerative facility. With the exception of the sun gear, all gears exhibited a tooth pattern that indicated full face width engagement. The sun-to-pinion mesh appeared to be end loaded. This may have been the result of a helix error ground into the sun gear, bending of the pinion shaft, or tilting of the pinion shaft in a tangential plane. The reason for the end loading can only be assessed if sun gear and pinions are inspected in detail and load sharing and vibration levels for the pinions are determined. Because load sharing was not measured dynamically in any of the testing, no statement can be made now of the presence of this phenomenon.

End-Loaded Gear Problem

One plausible explanation of the end-loaded sun pinion mesh may be lack of machining tolerance tight enough to prevent tilting of the pinion shaft in the tangential plane.

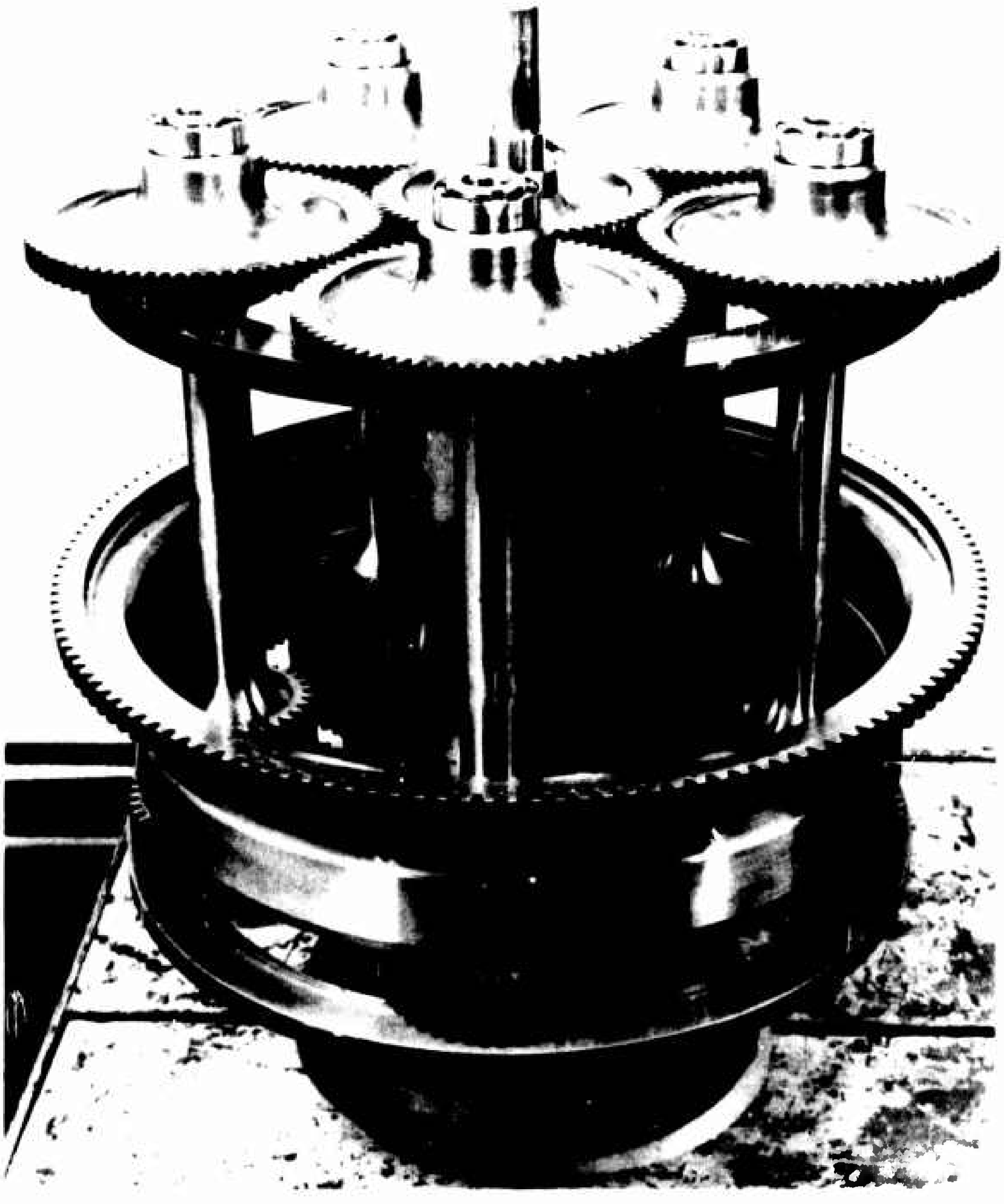


Figure 4. Curtiss Wright FP501 Test Unit

Figure 5 shows the magnified effect of loose indexing tolerance. The indexing tolerances of the FP501 free planet test unit (pinion-to-pinion indexing tolerance of the reaction and output ring gear meshes) were ±.003 inch.

Sikorsky Test Results

Another Eustis Directorate program, Roller Gear Drive Development Testing, had a similar indexing problem. Initial testing with roller gear drive was accomplished with parts held to ±.0002 inch. The final tolerance was ±.001 inch. Load sharing was measured dynamically, and the conclusion was drawn that an indexing tolerance greater than ±.003 inch is excessive. On the basis of the conclusions drawn by Curtiss-Wright in Design and Development Testing of Free Planet Transmission Concept, one can agree that the free planet transmission is a promising concept, but more testing is needed with respect to the statement that "the force balance principle . . . appears to be sound . . . dynamically." (Reference 2).

Frosted Zone Problem

Another potential problem uncovered during examination of the Curtiss-Wright FP501 was a waviness, or frosted zone, on the pinion bearing journal diameters. They are shown in Figure 6.

The frosted zone may be the result of (1) vibration of the pinion in a tangential plane due to poor indexing tolerances, (2) lack of roundness in grinding bearing journal diameters during manufacture, or (3) skidding during various drive conditions of the rings on the pinion bearing journal diameters. Frosted zones can be shown to be a minor problem. Whatever the initial cause, they do not necessarily cause stress concentrations large enough to make the surface distress self-propagating. In fact, a smoothing over and plastic spreading occur on the higher unpeeled surface. Figure 7 is an excellent example from the Roller Gear Drive R&M Test. Pinion S/N33 was used in a bench test and completed 200 hours of testing. Examination revealed a frosted zone on the lower roller. This pinion was then used in the R&M test. Examination at the end of 22.5 hours of R&M testing revealed that the frosted zones and peeling evident after the 200-hour test had disappeared. This self-healing phenomenon was noted earlier by Franklin Institute Research Laboratories in "Derivation of a Fatigue Life Model for Gears," USAAMRDL TR-72-14, which observed that shallow spalling of rolling contacting elements did not propagate deeper. The report concluded, "...a form of compliance may be responsible for the fact that cracks at the bottom of the shallow spall did not propagate under the Hertzian stresses or from lubricant-induced hydraulic pressure propagation."

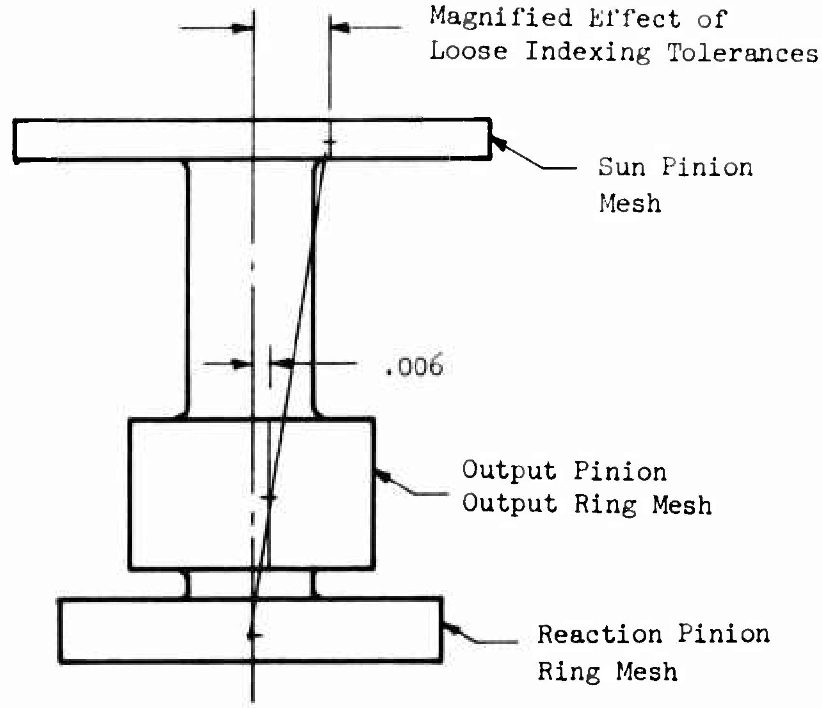


Figure 5. Pinion Shaft with Magnified Indexing Tolerances

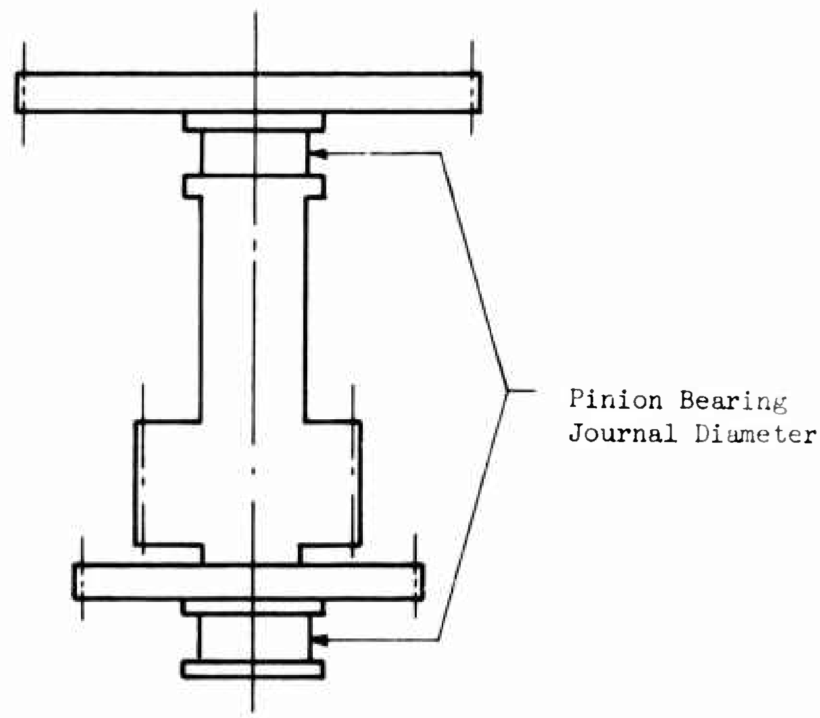


Figure 6. Schematic Pinion Shaft with Pinion Bearing Journal Diameters

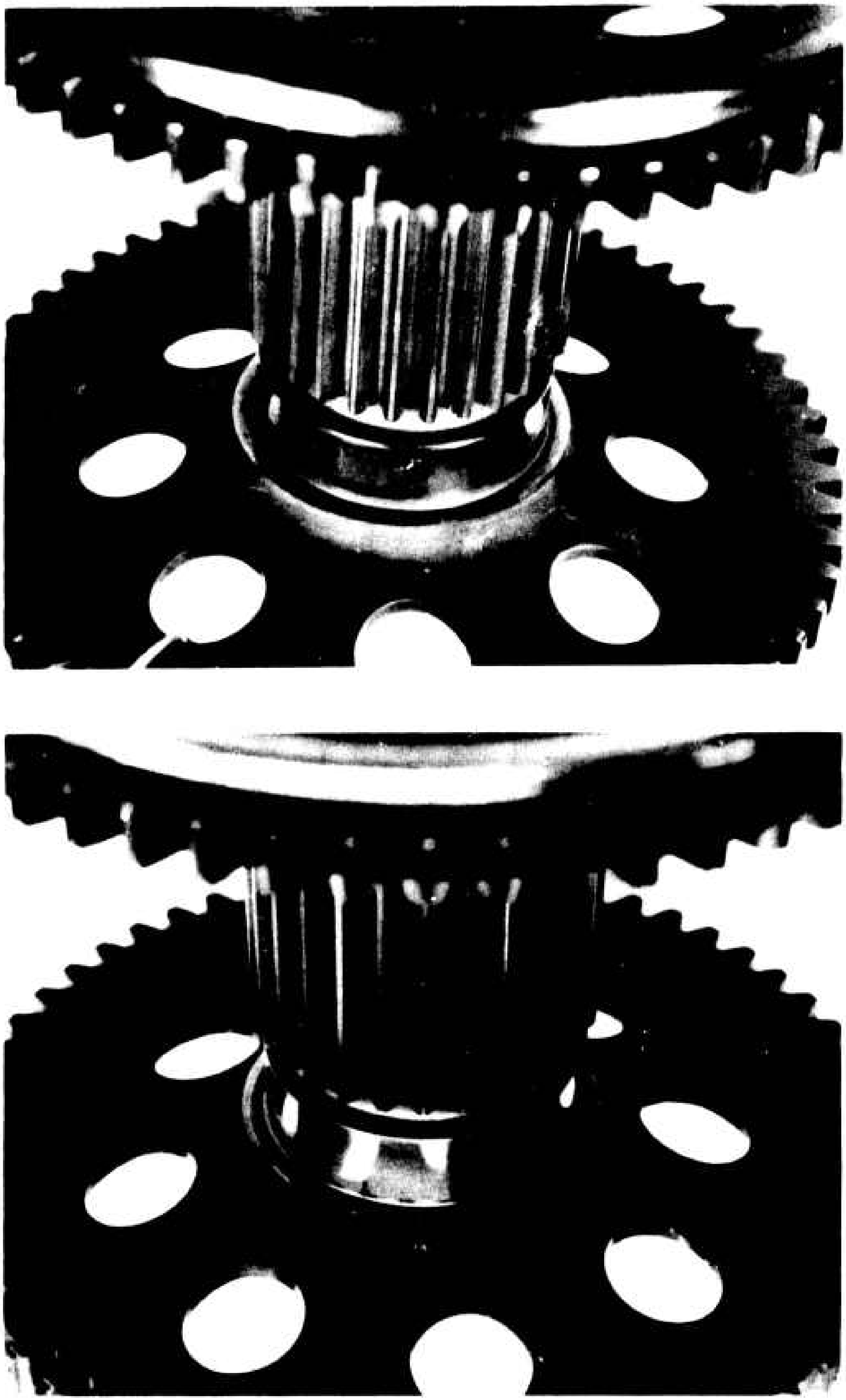


Figure 7. Roller Gear Drive First-Row Pinions

TYPES AND ARRANGEMENTS OF FREE PLANET DRIVES

The free planet drive can take many forms and arrangements. Each type has its own advantages and disadvantages. Schematics of some of the various types of free planets are shown in Figures 8 through 10. The possibilities of designing free planets have no theoretical limitation, but only two possible configurations appear practical. One is the power hinge used for flap actuation, and the other is the three-gear pinion free planet design, which is the subject of this study.

Power Hinge

Initial work in the design of free planet transmissions was performed with the power hinge, which is similar to the free planet transmission design selected for the present study. To achieve equilibrium, forces and moments must add up to zero about any point in or parallel with three planes. This is illustrated in Figure 1 for the forces acting on the power hinge. In a radial plane, the radial separating forces resulting from action of the gear teeth are reacted by free-floating rings. A force balance in the radial plane indicates

$$F_{e_{rad}} + F_{c_{rad}} - F_{s_{rad}} = 2F_{sup}$$

The use of two output ring gears in symmetry prevents the planets from skewing and, as shown, the summation of moments about any point in the radial plane is zero.

In the transverse plane of Figure 1, the sum of the forces is zero, since

$$F_c + F_s - F_e = 0$$

The moment in the transverse plane to produce the desired reduction ratio is also zero, since

$$F_s\, s + F_c\, x - F_e\,(xz + x) = 0$$

It is also clear that forces and moments acting in parallel with the tangential plane are also balanced by this symmetrical arrangement.

Other Types

Other types of free planet drives are reported in References 2 and 3.

Figures 8 through 10 show some of the possible free planet drive concepts. All would work, but they appear to be unnecessarily complicated. The unsymmetrical free planet design shown in Figure 8 was advanced in the Advanced Technology VTOL Drive Train Configuration Study. It is a possible solution, but no hardware has been built to demonstrate this concept.

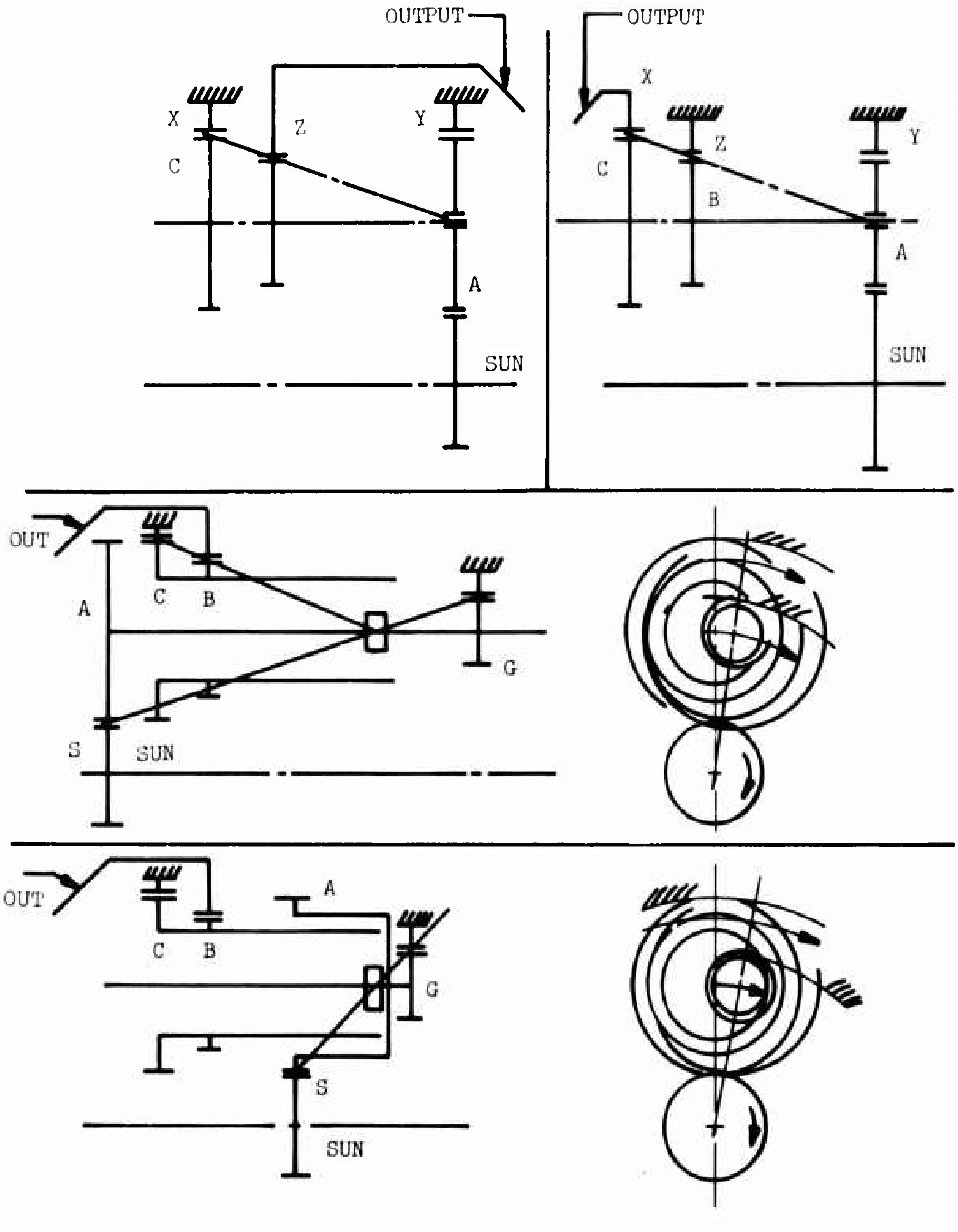


Figure 8. Free Planet Schematics

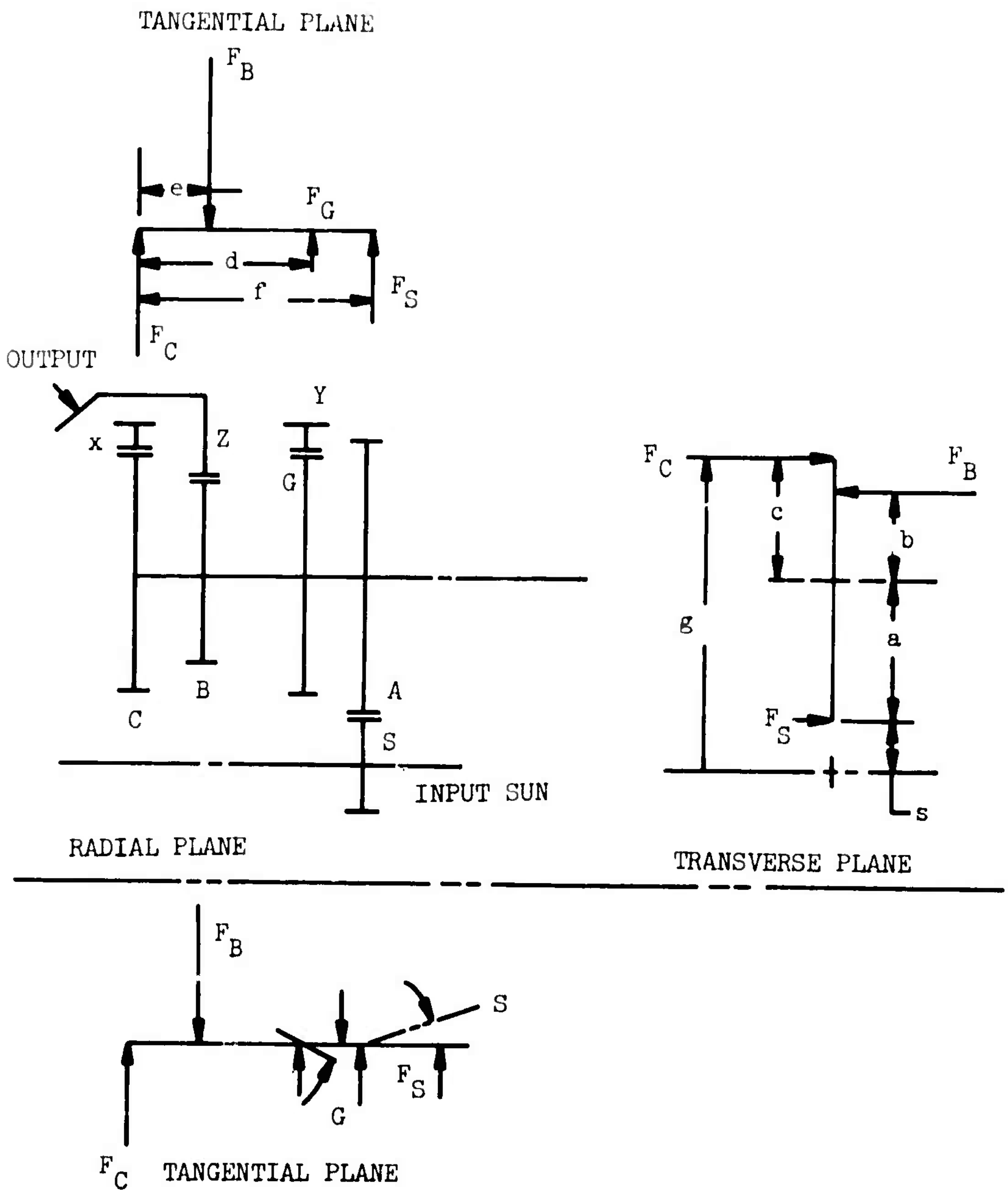


Figure 9. Free Planet Schematics - Four Load

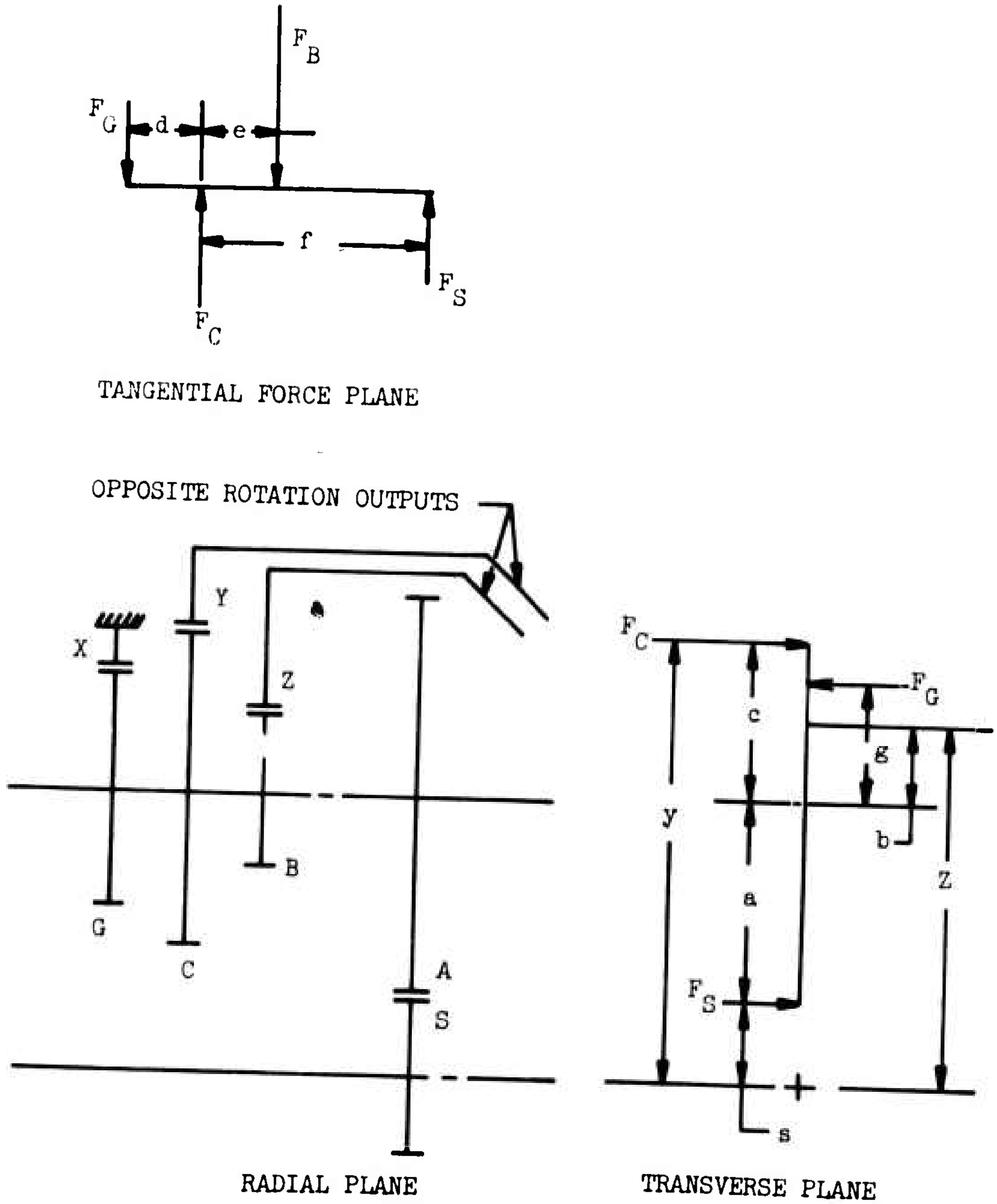


Figure 10. Free Planet Schematic - Dual Rotation Outputs With Plane Forces .

Three Gear Pinion Free Planet

The free planet drive shown in Figure 11, which is the subject of this report, consists of a sun gear input, free planet pinion with three spur gears per shaft, output ring gear on central free planet pinion, and fixed ring gear on outer free planet pinion. This type of arrangement for a helicopter drive offers a reduction ratio range from 5-to-1 up to 30-to-1. The force balance criterion is much the same as in the power hinge example discussed previously.

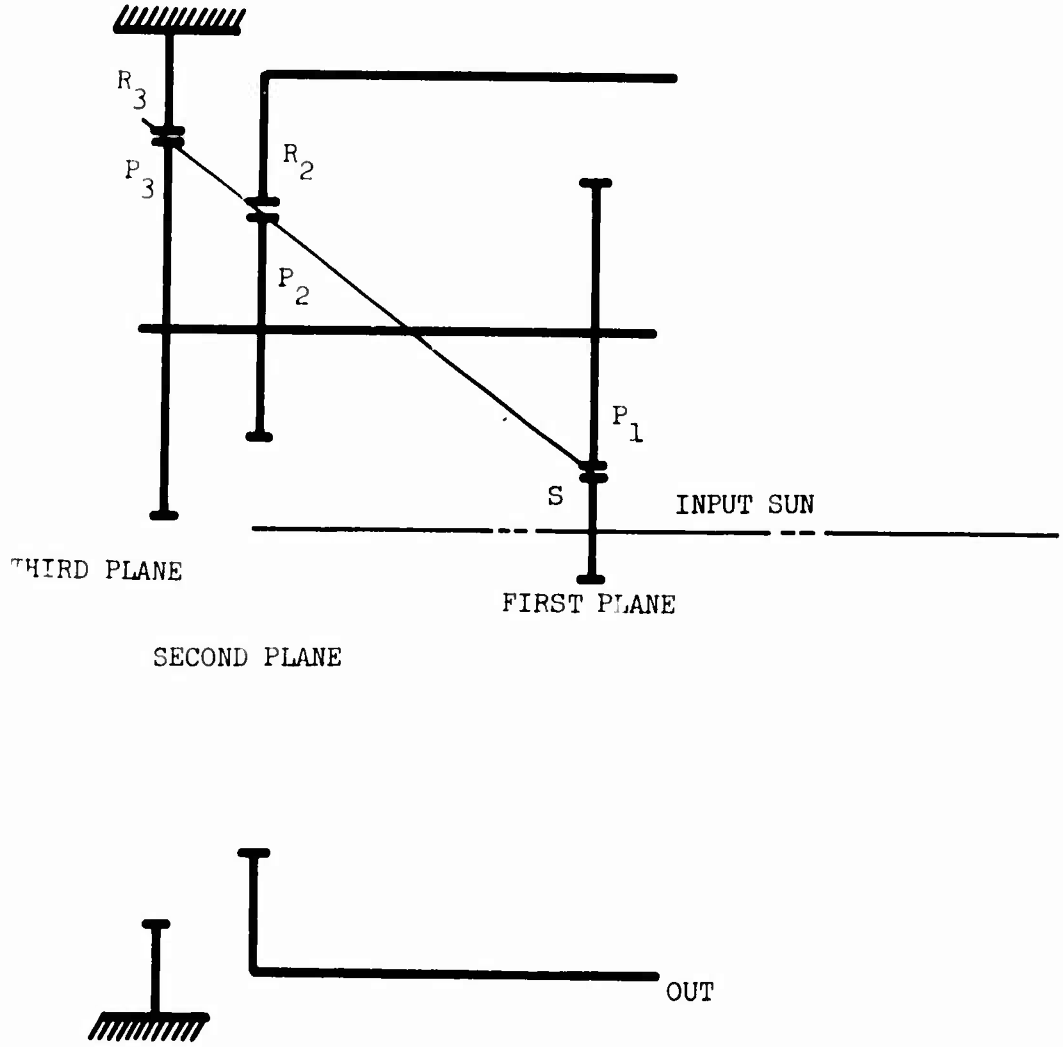


Figure 11. Three Gear-Free Planet Schematics

PRELIMINARY DESIGN

Design Requirements

The baseline aircraft is a medium-sized utility transport (MUT), the same as that used in the Advanced Helicopter Structural Design Investigation Study. This baseline aircraft design uses UTTAS technology and provides internal volume for crew, litters, passengers, cargo, estimated fuel, and equipment. Table 1 lists MUT baseline aircraft data. Figure 12 shows the aircraft general arrangement. Table 2 lists the speed and maximum horsepower design requirements for the free planet transmission.

The transmission gears and shafts are designed for infinite life. Bearings are designed for 3,000 hours B.10 life minimum at the power and speeds listed in Table 34. Accessory drives are located in the rear cover of the main transmission.

Envelope, Rotation, and Ratio Restrictions

The paper engines for the baseline MUT had an output speed of 30,000 rpm and develop 925 HP per engine. Since the MUT was designed for a main rotor speed of 340 rpm, an overall reduction ratio of 98.6:1 was required.

With the location of the engines and overall reduction ratio established, the number of reduction stages needed to deliver power from the engines to the main rotor shaft was examined. The fewer the reduction stages, the lighter the weight. Three configurations were examined that would deliver power with a minimum number of reduction stages.

Design Envelope Limits

At the start of preliminary design, limits were established for the transmission envelope. Since the main rotor shaft must pass through the center of the sun gear, the minimum possible diameter was set at 6.0 inches. The maximum ring gear diameter was established at 31.0 inches because of the size limitation of the quench press used during case hardening of the gear teeth. The free planet design parameters established are listed in Table 3.

In general, the lightest transmission will result when the highest possible reduction ratio is located in the final reduction stage. Therefore, the earlier reduction stages should have reduction ratios as low as possible.

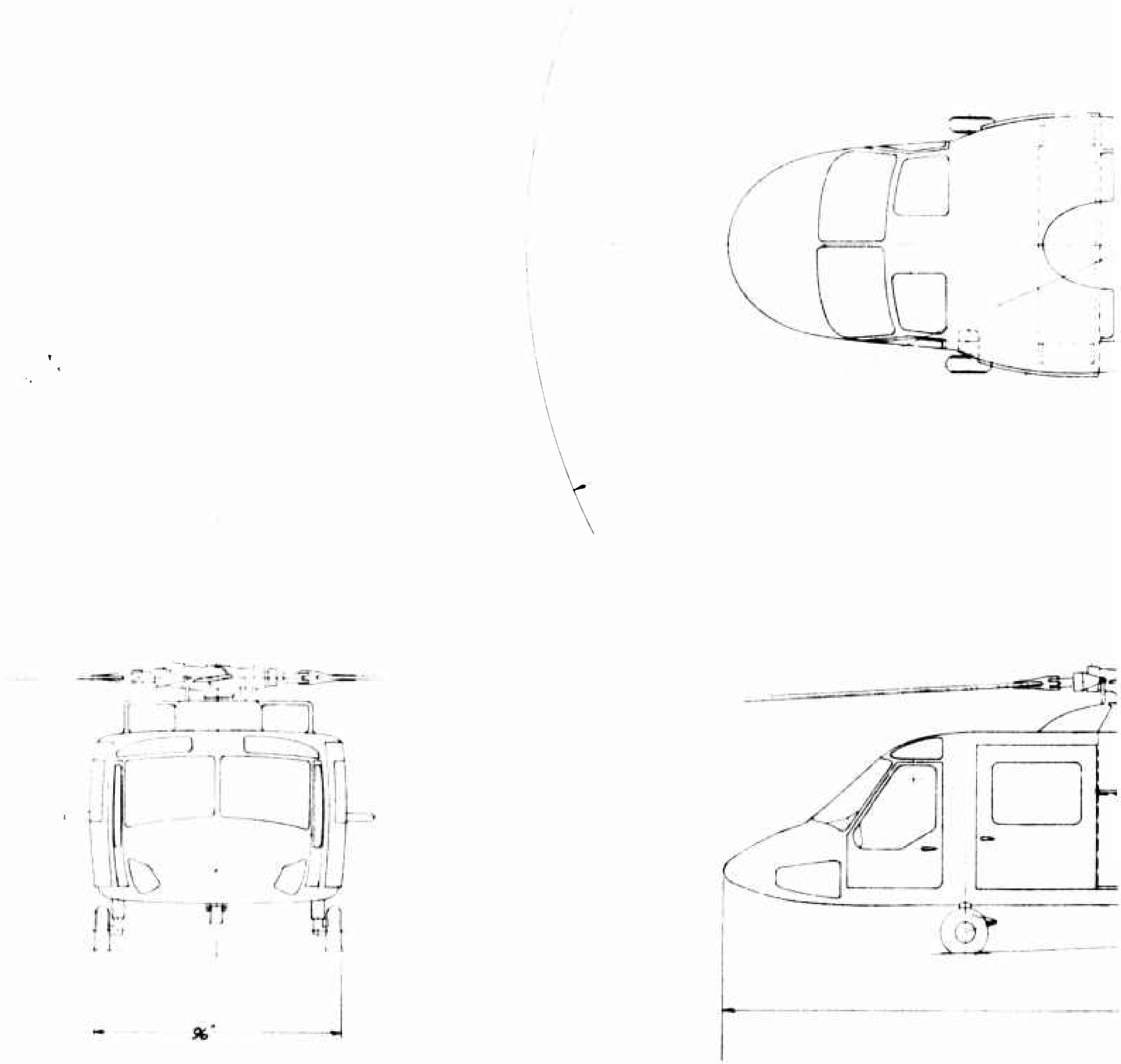
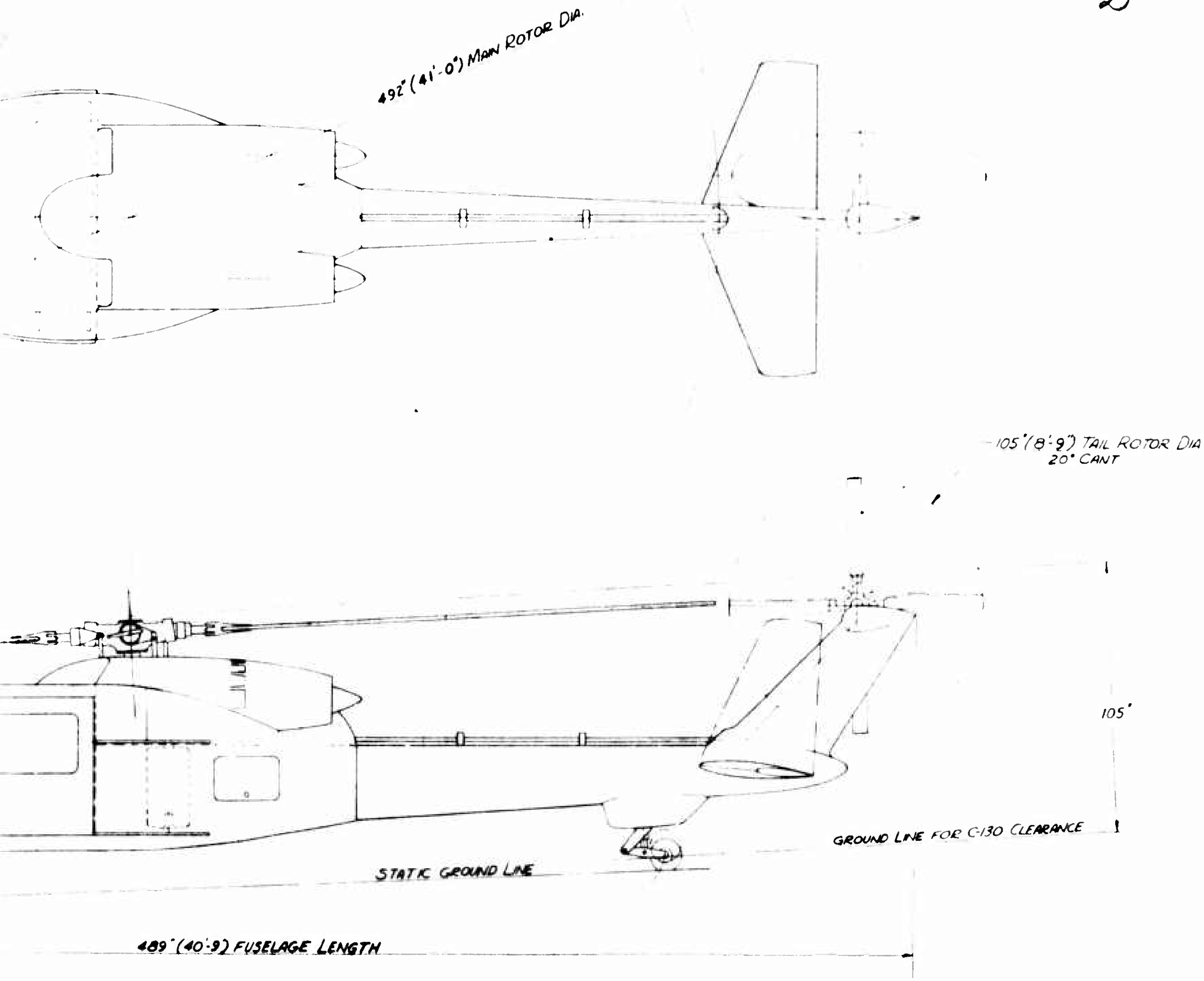


Figure 12. MUT Aircraft General Arrangement



492" (41'-0") MAIN ROTOR DIA.
105" (8'-9") TAIL ROTOR DIA
20° CANT
105"
GROUND LINE FOR C-130 CLEARANCE
STATIC GROUND LINE
489" (40'-9) FUSELAGE LENGTH

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TABLE 1. MUT BASELINE DATA SHEET

DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G W (LB)	9471.0	RADIUS (FT)	20.50	RADIUS (FT)	4.40
PAYLOAD (LB)	960.0	CHORD (FT)	1.322	CHORD (FT)	.535
WEIGHT EMPTY (LB)	6618.0	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (LB)	1389.0	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1178.0	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	700.0
HOVER + CLIMB HP	1261.0	ASPECT RATIO	15.511	ASPECT RATIO	8.231
MAIN ROTOR DESIGN HP	1048.0	CT/SIGMA	.0850	CT/SIGMA	.1089
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9239.4	TAIL ROTOR LIFT	231.9
M R DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7147
MAIN G B DESIGN HP	1564.0	BLADE AREA (SQ FT)	108.4	BLADE AREA (SQ FT)	9.4

TABLE 2. FREE PLANET TRANSMISSION DESIGN REQUIREMENTS

Location	Speed (rpm)	Power (hp max)
Input Drives		
Dual Engine	30,000	1,850
Single Engine	30,000	925
Main Rotor	340	1,454
Tail Takeoff Total		170
Tail Rotor Takeoff		120
Accessory Drives		
Generator (Two)	8,100	30
Tachometer Generator	3,900	1
Servo Hydraulic Pump	4,200	4
Aux Servo Hydraulic Pump	4,000	4
Utility Hydraulic Pump	4,200	7
Lubrication Pump	6,000	1

TABLE 3. FREE PLANET TRANSMISSION DESIGN PARAMETERS

Minimum Sun Gear Diameter	6.00 inches
Maximum Ring Gear Diameter	32.0 inches
Output RPM	340
Input RPM	12,000, - 15,000
Reduction Ratio	10 to 20:1
Gear Allowable Compressive Stress*	130,000 psi
Gear Allowable Bending Stress (One Way)	55,000 psi
Minimum Bearing Life (B.10 Life)	3,000 hours
Roller Allowable Compressive Stress	150,000 psi

* Using AGMA calculation method

TRANSMISSION LAYOUT ARRANGEMENTS FOR MEDIUM UTILITY TRANSPORT

Dual High-Speed Bevel Gear Inputs

The first arrangement, shown in Figure 13, employs a freewheel unit driven by the engine, whose output drives a bevel gear set. The output bevel gear of this set, which is concentric with the main rotor shaft, is the combining gear for both engines. It drives the final reduction stage and the tail takeoff. The final stage is a free planet reduction unit with output to the main rotor shaft.

The use of two engines on the MUT in a vee arrangement creates the problem of large overall width of the aircraft, which impacts on air transportability. The excessive width problem will be further aggravated in the future by the addition of IR suppressors.

If the aircraft required a single-engine arrangement, this bevel gear input drive would offer the lightest weight, smallest number of parts, and full use of the high ratio capability of the free planet drive unit. A schematic of such an arrangement is shown as Figure 14.

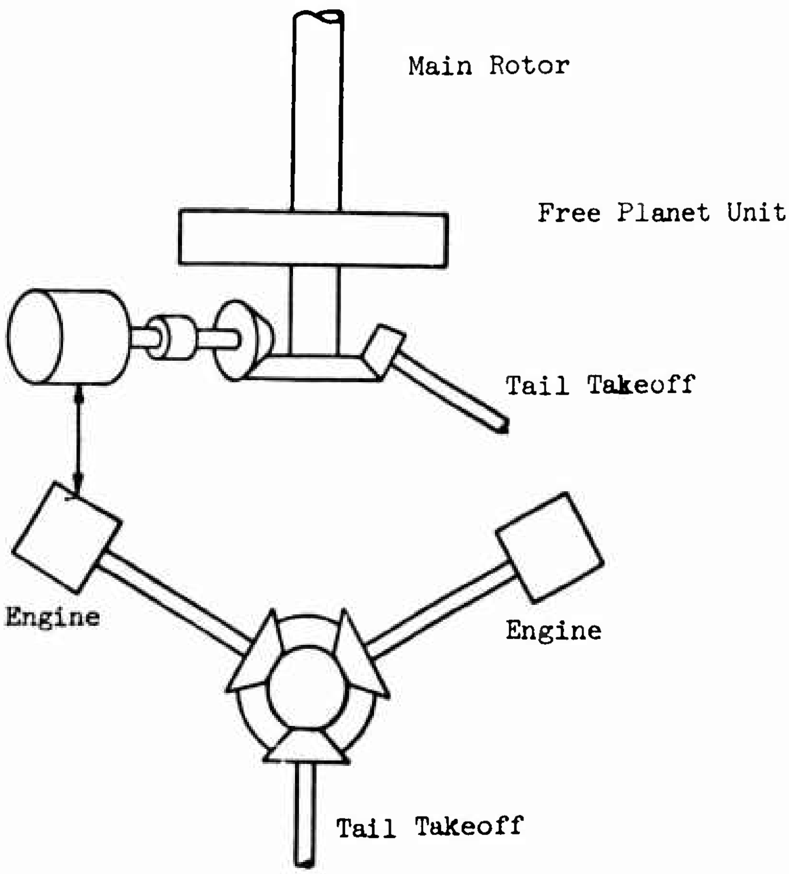


Figure 13. Dual High-Speed Bevel Gear Inputs

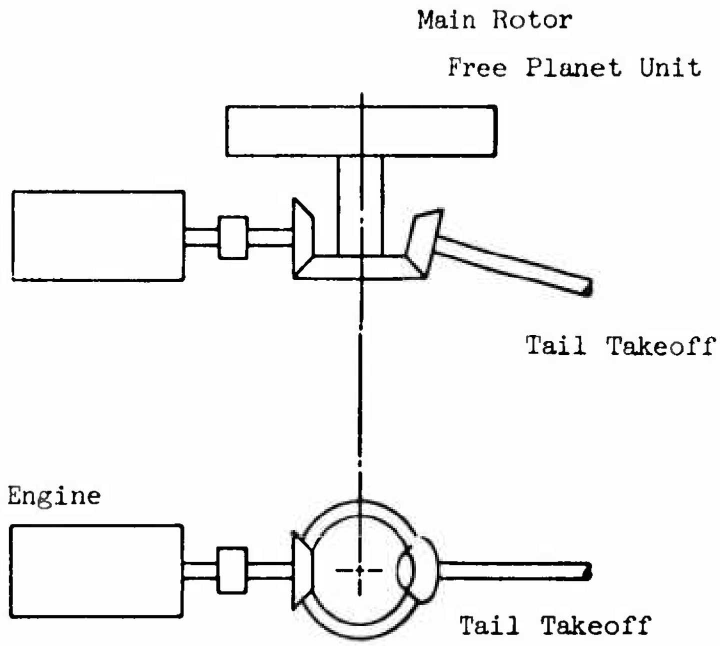


Figure 14. Single High-Speed Bevel Gear Input

Dual High-Speed Crossed Helical Gear Inputs

The second configuration considered is shown in Figure 15. The first stage of this system is a crossed helical mesh, which permits parallel engine mounting and a wide center distance between engines. The driven helical gear transmits power through a freewheel unit to the second-stage combining spiral bevel mesh. The driven gear of the spiral bevel mesh is concentric with the main rotor shaft and drives both the tail takeoff and free planet reduction unit. This configuration was rejected, because crossed helical gears are inefficient for high-power, high-torque applications.

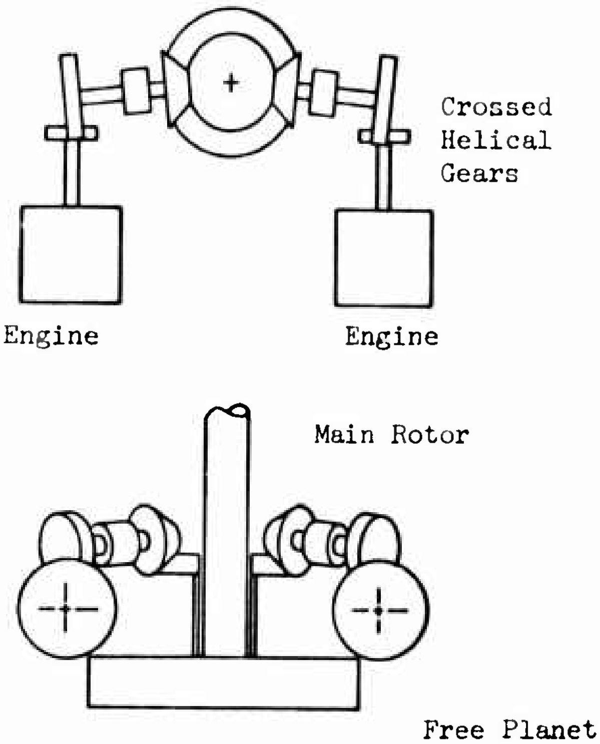


Figure 15. Dual High-Speed Crossed Helical Gear Input

Dual High-Speed Spur Gear Inputs

The third possibility is shown in Figure 16. In this configuration, the engine drives through a freewheel unit to the first stage combining spur gear mesh. The output gear of the spur gear mesh drives a single spiral bevel mesh which turns the corner. The output bevel gear is concentric with the main rotor shaft and drives the tail takeoff and free planet reduction unit.

This design was selected, since the bearing problem on a high-speed 30,000-rpm spur mesh is much easier to solve than on a 30,000-rpm bevel mesh. Through the use of idler gears, the necessary spacing is obtained between the engines, and pads are provided for an accessory drive.

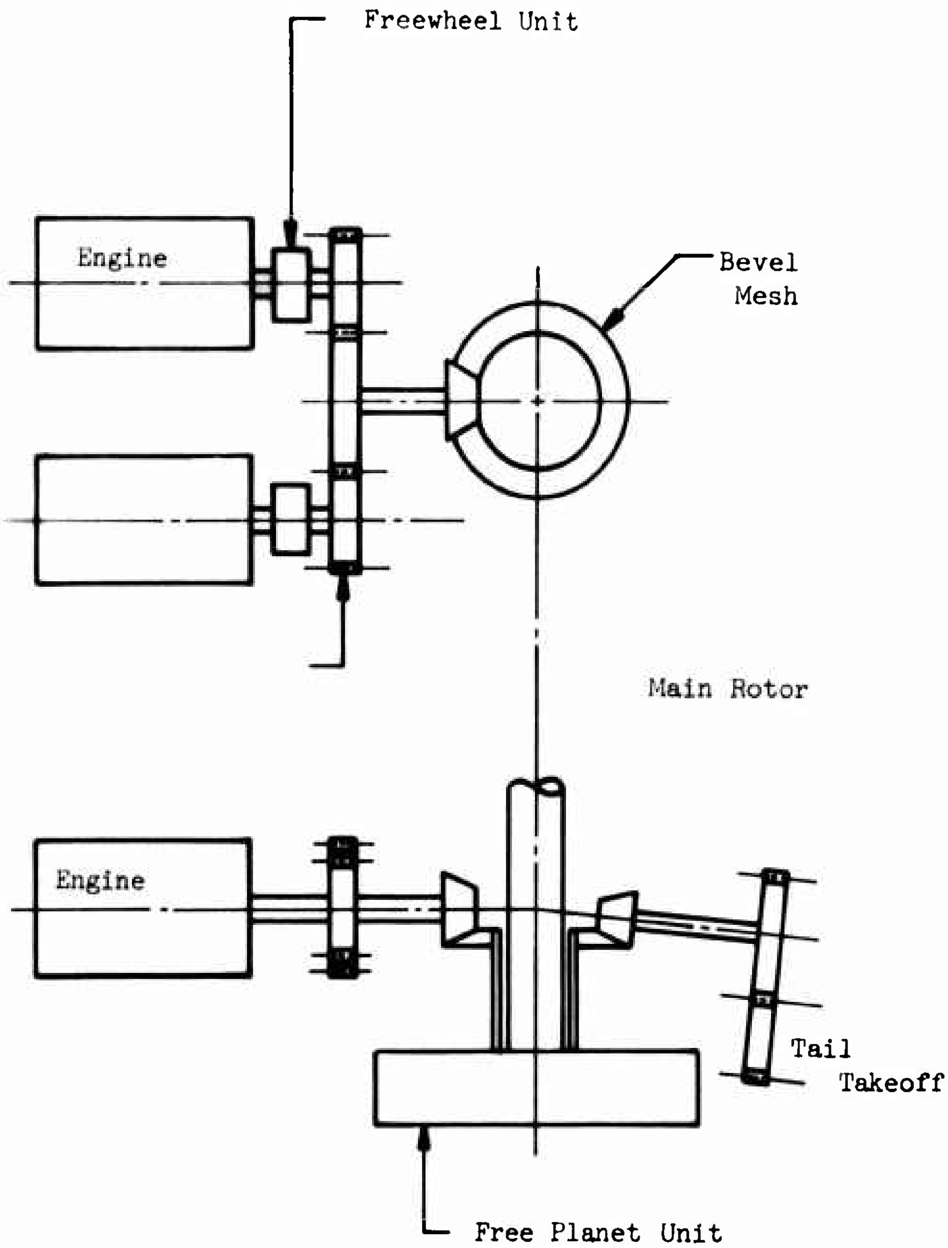


Figure 16. Dual High-Speed Spur Gear Inputs

FREE PLANET TRANSMISSION CONCEPT FOR UTTAS-SIZE AIRCRAFT

A preliminary design study of the application of the free planet concept to a UTTAS-size twin turbine helicopter indicates that the concept is unsuitable, primarily because the high reduction ratio it offers cannot be fully used without significant compromises in engine location.

To take full advantage of the free planet concept, only one other stage of gearing is needed between the engine and the main rotor. For a twin-engine helicopter using front-drive engines, one possible solution is to locate the engines horizontally aft of the main gearbox in a vee configuration, so their input bevel pinions mesh with a common bevel gear. The fundamental objection to this arrangement is the effect on aircraft balance, since the engines must be located farther forward than on a smaller aircraft, such as MUT or ASH. This situation is expected to be aggravated in the future by the addition of IR suppressors. When additional factors such as engine inlet ducting, easy accessibility, and ballistic survivability are considered, the present UTTAS engine arrangement which requires two bevel gear stages is hard to beat. When the overall UTTAS reduction ratio is spread over three stages of speed reduction to meet the geometry constraints, only a simple planetary is required for the output stage.

FREE PLANET UNIT DESIGN

To select a free planet unit to meet the constraints of the MUT aircraft, consideration has to be given to reduction ratio, gear teeth stress, axial length, pinion shaft design, and roller ring design. Evaluation of the interactive effects of these attributes and simplification of the selection process made it necessary to develop a computer program, shown in Appendix A.

Reduction Ratio

The reduction ratio for the free planet is determined through use of the equivalent system method. The equivalent system and the free planet configuration are shown in Figure 17. The equivalent system has the same relative pitch-line velocities as the actual system, and all gears rotate

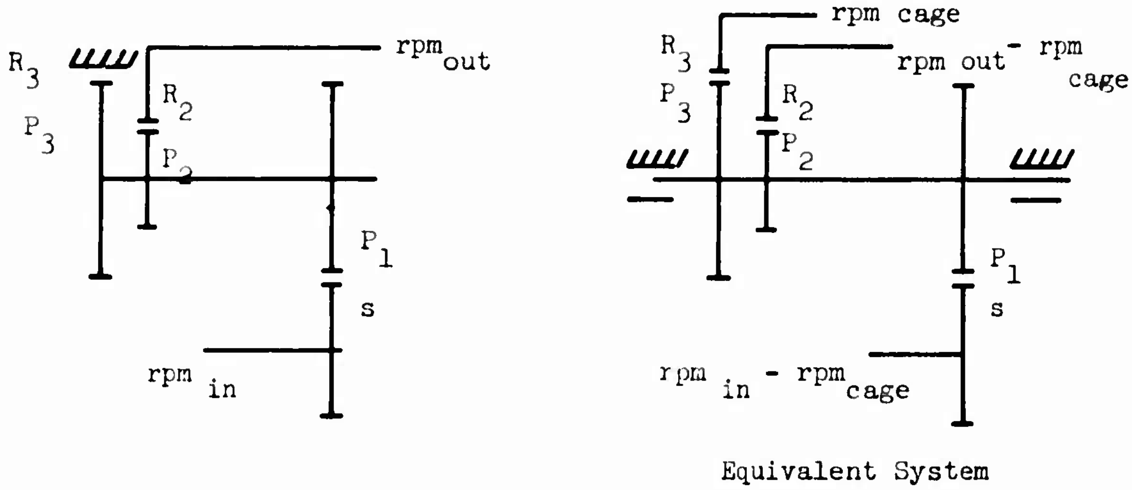


Figure 17. Actual and Equivalent Free Planet Systems

on fixed centers. This simplifies the task of determining the reduction ratio of the system. A minus sign indicates that the member turns in the opposite direction from the input. In the equivalent system, all gears are on fixed center. The speed of rotation of any shaft can be found in terms of the speed of rotation of any other shaft:

$$-rpm_c \frac{R_3}{P_3} = (rpm_{out} - rpm_{cage})\left\{\frac{R_2}{P_2}\right\} = (rpm_{in} - rpm_{cage})\left\{-\frac{S}{P_1}\right\}$$

Simplifying and solving for rpm_{in}/rpm_{out} results in

$$RR = \frac{rpm_{in}}{rpm_{out}} = \frac{\left\{1 + \frac{R_3P_1}{P_3S}\right\}}{\left\{1 - \frac{R_3P_2}{P_3R_2}\right\}}$$

Gear Tooth Stress Analysis

The dynamic bending stresses and compressive stresses for the gear teeth of the drive train are calculated and compared with an allowable stress.

Bending Stress Equation

The basic equation for determining the bending stress at the root of a tooth in a spur or bevel gear is as follows:

$$f_b = \frac{W_t K_o}{K_v} \cdot \frac{P_d}{F} \cdot \frac{K_s K_m}{J}$$

where

W_t = tangential tooth load

K_o = overload factor

K_v = dynamic factor

P_d = diametral pitch

F = face width

K_s = size factor

K_m = load distribution factor

J = geometry factor

All the free planet transmission drive gears are case carburized and ground to close tolerances to minimize dynamic effects. The dynamic factor as a result is, therefore, taken as 1.0.

The overload factor makes allowances for the roughness or smoothness of operation of the driving and driven members. Again, this factor is taken as 1.0.

The load distribution factor accounts for the combined effects of deflection of mountings and misalignment of gears. For bevel gears, the load distribution factor is less critical and is taken as 1.10. For the free planet, which is considered to be less critical than a planetary drive, the load distribution factor is taken as 1.10. The load distribution factor for conventional spur or helical gears is taken to be 1.30.

The size factor reflects nonuniformity of material properties and is taken as 1.0 for aircraft spur gears. For bevel gears, the size factor is a function of the diametral pitch.

The geometry factor evaluates the shape of the tooth, the position at which the most damaging load is applied, stress concentration due to geometric shape, and load sharing. In bevel gears the geometry factor is taken for the mean normal section of the tooth.

Compressive Stress Equation

The contact stress for steel spur gears can be calculated by

$$f_c = \sqrt{\frac{(21)(10^6)\ W_t}{(\mathrm{Sin}\ 2\ )(F)} \left\{ \frac{1}{d_p} \pm \frac{1}{d_s} \right\}} \qquad \begin{array}{l} + \text{ for external} \\ - \text{ for internal} \end{array}$$

For bevel gears, the contact stress is given by

$$f_c = K_p \sqrt{\frac{2T_p\ K_o}{K_v} \cdot \frac{1}{F\ d_p^{\ 2}} \cdot \frac{K_s\ K_m\ K_f}{I}}$$

where

K_p = elastic coefficient

K_o = overload factor

K_v = dynamic factor

T_p = pinion torque

d_p = pinion pitch diameter

K_s = size factor

K_m = load distribution factor

K_f = surface condition factor

I = geometry factor

Allowable Stresses

Table 4 gives the allowable stresses for carburized and ground steel.

The difference in allowable stresses for spur and bevel gears is due mainly to the different size factors used.

TABLE 4. ALLOWABLE GEAR BENDING AND CONTACT STRESSES

Components	Allowable Stress
Spur Gears - One-Way Bending	F_b = 55,000
Bevel Gears - One-Way Bending	F_b = 30,000
Spur Gears	F_c = 130,000
Bevel Gears	F_c = 200,000

Axial Length Determination

One necessary condition for the free planet transmission is that the sum of the moments about any point in a radial plane equal zero. This can be accomplished simply during design selection by spacing the pinion shaft gears so that they lie along the balance line shown in Figure 18.

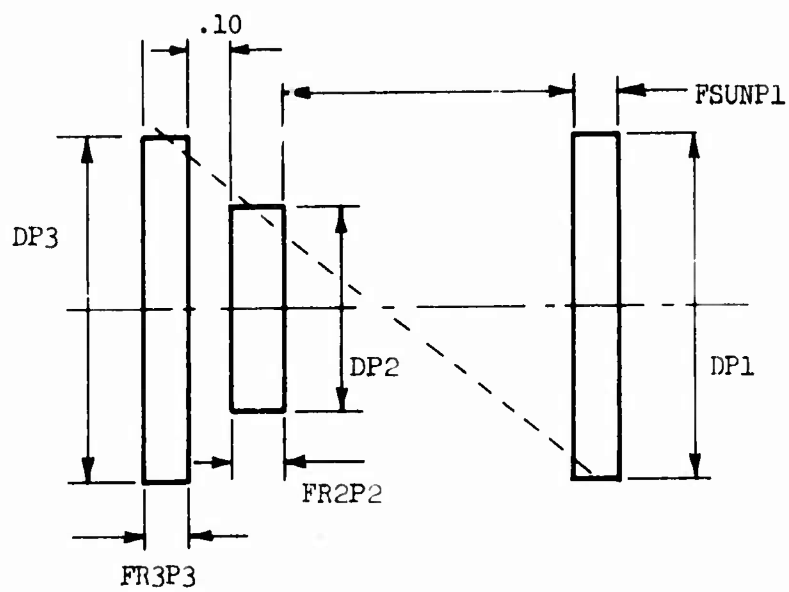


Figure 18. Axial Length Pinion Shaft Schematic

The geometry is shown in Figure 19.

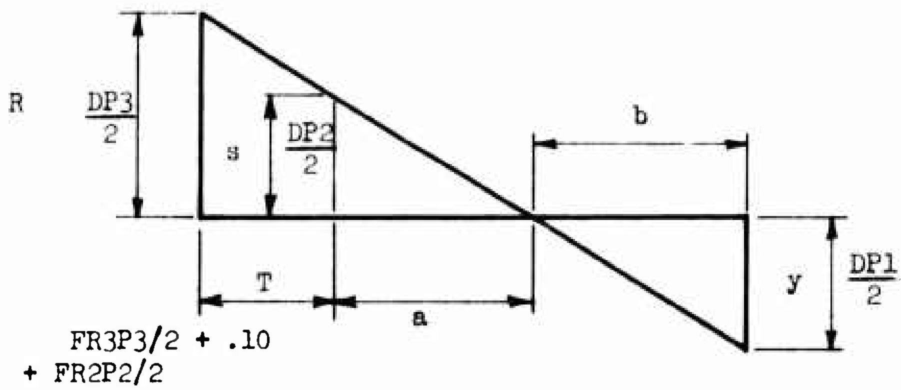


Figure 19. Axial Length Pinion Shaft Geometry

$$\frac{a}{s} = \frac{a + T}{R}$$

$$\frac{b}{y} = \frac{a + T}{R}$$

$$a = \frac{st}{R - s}$$

$$b = y \, \frac{a + T}{R} = y \, \frac{\frac{ST}{R - S} + T}{R}$$

$$\text{Axial} = T + a + b + \frac{FR3P3}{2} + \frac{FSUNP1}{2}$$

$$= FR3P3 + \frac{FR2P2}{2} + .10 + \frac{DP2}{2(DP3 - DP2)} (FR3P3 + .20 + FR2P2)$$

$$+ \frac{DP1}{2(DP3 - DP2)} (FR3P3 + .20 + FR2P2) + \frac{FSUNP1}{2}$$

Pinion Shaft Design

The pinion shaft of the free planet transmission is designed to carry both torque and bending. The torque in the pinion shaft is the result of the differences in torque of the reaction ring gear, output ring gear, and sun gear torques. The resulting loads on the pinion shaft in both the tangential and transverse directions are shown in Figure 20.

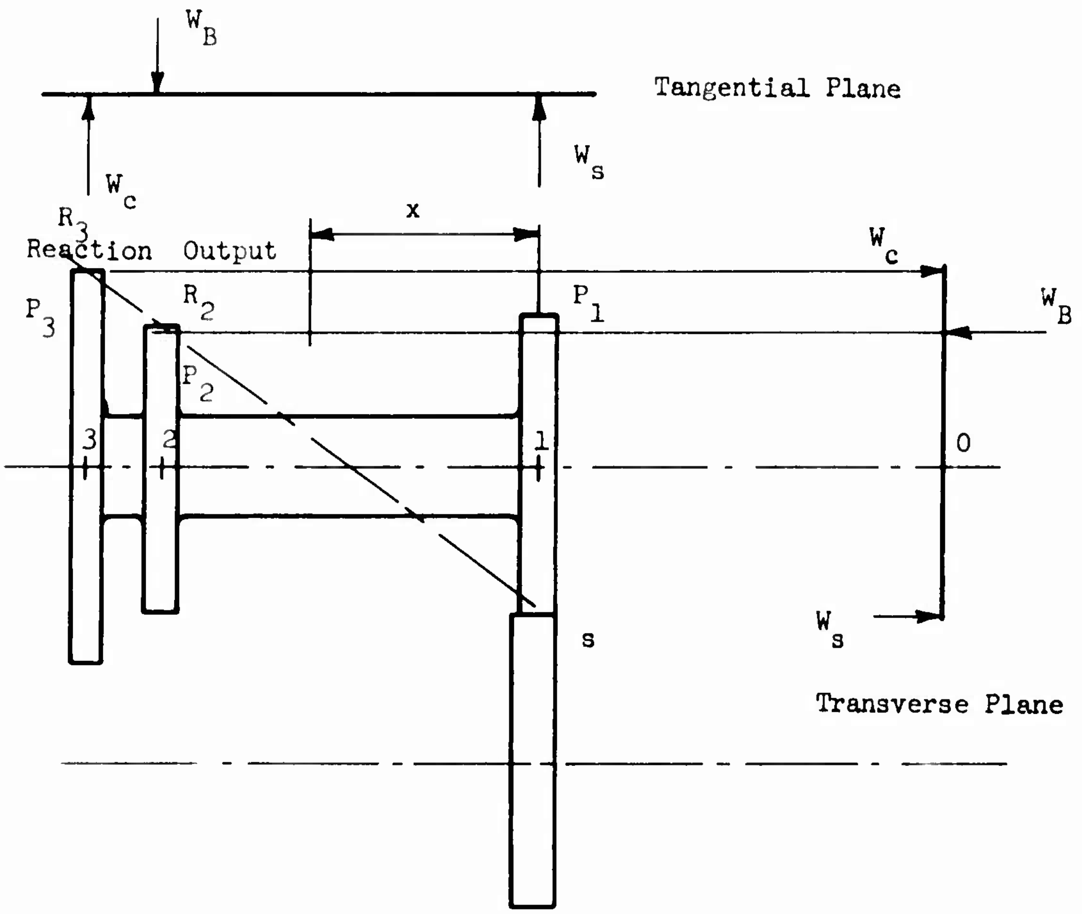


Figure 20. Pinion Shaft Loading

The torque between points 1 and 2 is given by

$$T_{1-2} = W_s \left\{\frac{DP_1}{2}\right\}$$

$$T_{2-3} = W_c \left\{\frac{DP_3}{2}\right\}$$

From a torque balance in the transverse plane, it can also be shown that

$$W_c \left\{\frac{DP_3}{2}\right\} = W_a \left\{\frac{DP_2}{2}\right\} + W_s \left\{\frac{DP_1}{2}\right\}$$

In the tangential plane, the bending moment on the pinion between points 1 and 2 is equal to

$$M_{1-2} = W_s\, x$$

The maximum moment on the pinion shaft between points 1 and 2 is equal to

$$M_{\substack{1-2\\MAX}} = W_s \left(a + b - \frac{FR2P2}{2}\right)$$

To design for a fatigue design condition, the torque is considered steady, and the moment is vibratory and subject to complete reversal.

$$f_{\substack{vib\\1-2}} = \frac{M_{\substack{1-2\\MAX}}}{Z}$$

$$f_s = \frac{T_{1-2}}{Z_p} = \frac{T_{1-2}}{2Z}$$

As shown in Reference 4, the margin of safety using the maximum shearing stress theory of failure is equal to

$$MS = \frac{1}{\sqrt{\left\{\frac{K_t f_v}{Fen}\right\}^2 + 4\left\{\frac{f_s}{F_{ty}}\right\}^2}} - 1 = 0$$

$$f_{\substack{vib\\1-2}} = \frac{W_s \left(a + b - \frac{FR2P2}{2}\right)}{Z}$$

$$f_s = \frac{W_s\ DP_1}{(2)(2)(Z)}$$

For an open section, $K_t = 1.0$.

Since the pinion shaft is made of 9310 steel AMS 6265 with a core hardness of Rockwell C 30-45, the material properties are:

$$F_{tu} = 136{,}000$$

$$F_{ty} = 115{,}000$$

$$F_{en} = 44{,}500$$

$$SEF = .72$$

$$Pf = .70$$

$$\left\{\frac{K_t fv}{F_{en}}\right\}^2 + \left\{\frac{f_s}{F_{ty}}\right\}^2 = 1$$

$$\left\{\frac{W_s\left(a + b - \frac{FR2P2}{2}\right)}{22400\ Z}\right\}^2 + \left\{\frac{W_s DP_1}{(2)115000)(Z)}\right\}^2 = 1$$

$$Z = \left[\left\{\frac{W_s\ (a + b - \frac{FR2P2}{2})}{22400}\right\}^2 + \left\{\frac{W_s DP_1}{230{,}000}\right\}^2\right]^{\frac{1}{2}}$$

where

FR2P2 = face width of output ring pinion mesh

$$T = \frac{FR3P3}{2} + .10 + \frac{FR2P2}{2}$$

$$a = \frac{DP2(T)}{DP3 - DP2}$$

$$b = \frac{DP1}{DP3}\ (a + T)$$

$$W_s = \frac{T_s\ (2)}{(DS1)(Nopin)}$$

The flexibility of the pinion shaft or the wind-up is determined from

$$\varnothing = \frac{T}{Z_p G r_o} = \frac{T}{2ZGr_o}$$

The size of the pinion can be determined for the torsional and bending loads. In addition, the torsional wind-up can be determined.

Roller Ring Loads

Equilibrium was considered only in the tangential and transverse planes. To establish equilibrium in the radial plane, roller rings are required to react the centrifugal loads and the gear-separating loads. Figure 21 is a free body representation of the pinion shaft in the radial plane.

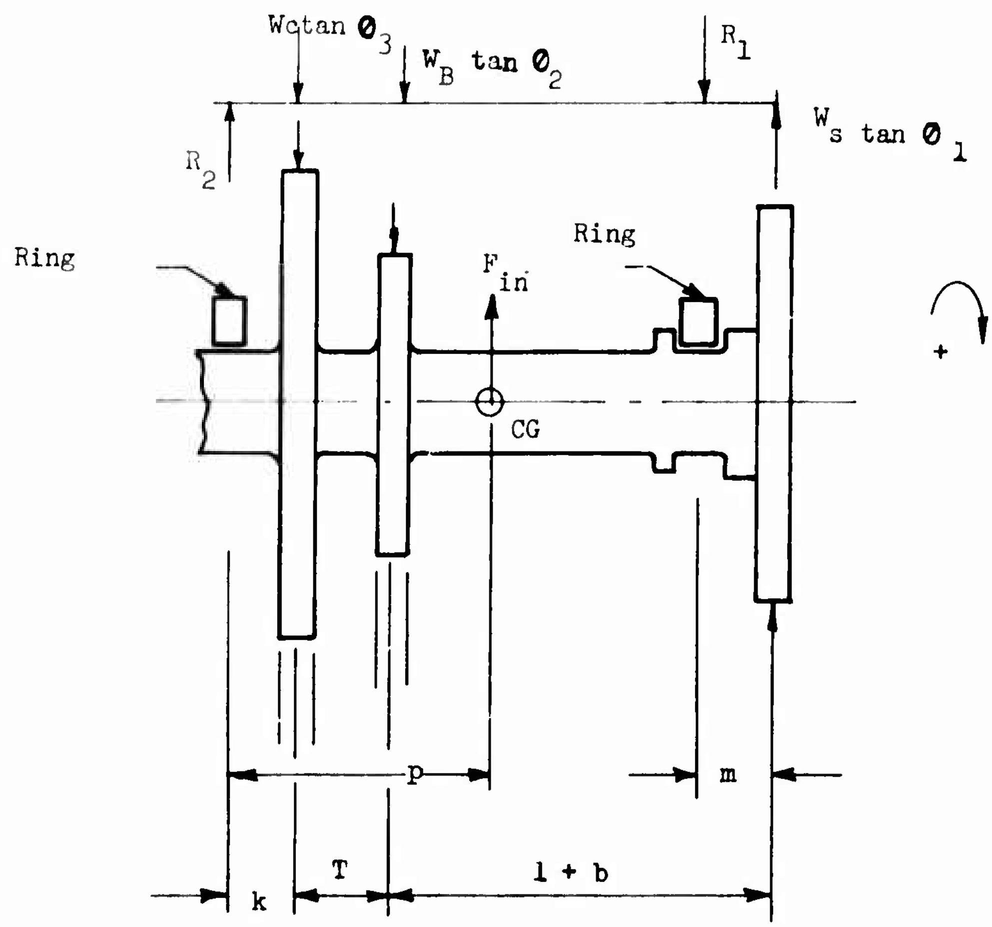


Figure 21. Roller Ring Loading

Summing the moments about point R_2,

$$W_c(\tan\emptyset_3)(K) + W_B(\tan\emptyset_2)(K + T) - F_{iner}P + R_1(K + T + a + b - m)$$

$$- W_s\tan\emptyset_1(K + T + a + b) = 0$$

$$F_{inertia} = mr^{\;\;2}$$

$$= \left(\frac{\text{Weight}}{32.2}\right)\left(\frac{\text{radius}}{12}\right)\left\{\left(\frac{2}{60}\right)\ \text{RPM}\right\}^2$$

Solving for R_1,

$$R_1 = \frac{1}{K + T + a + b - m}\left\{W_s\ \tan\emptyset_1(K + T + a + b)\right.$$

$$\left. + F_{iner}\ P - W_c\ K\ \tan\emptyset_3 - W_b(K + T)\ \tan\emptyset_2\right\}$$

Similarly, summing the moments about point R_1 and solving for R_2

$$R_2 = \frac{1}{K + T + a + b - m}\ \left\{W_c\ \tan\emptyset_3(T + a + b - m) + W_B\ \tan\emptyset_2(a + b - m)\right.$$

$$\left. + W_s\ \tan\emptyset_1 m - F_{inertia}(K + T + a + b - p - m)\right\}$$

For the case of a start-up condition or operation at close to zero speed, R_2 may be negative, requiring the addition of a roller on the inside roller diameters.

Free Planet Design Selection Model

A computer model was developed to aid in selection of a free planet unit for the baseline MUT. Appendix A is a listing of the actual program used. Figure 22 is a simplified flow chart of the program logic.

With the criteria already established, few acceptable designs were anticipated, but thousands were found suitable. Development of other criteria narrowed the selection. Table 5 is a list of the criteria used.

Figures 23 and 24 summarize some of the computer-generated results. Figure 23 is a map of possible free planet designs plotted on the basis of reduction ratio and overall unit height. Higher ratios lead to larger overall height, but this height increase can be limited by operating on what can be called the efficient frontier. Figure 24 illustrates the effect of changing the sizes of the sun gear and ring gears. The lowest weight is achieved with the lowest reduction ratio and lowest height. For a given reaction ring gear size and required reduction ratio, the smaller the sun gear, the lower the overall unit weight and height. Contrary to what might be expected for a given sun gear and reduction ratio, decreasing the reaction ring diameter increases gearbox height and weight.

Therefore, the flatter the free planet package, the lighter the overall weight. The decrease in weight continues until the size and weight of the roller ring diameter increase faster than the weight of the gearing decreases. The effect of increased roller ring weights was not significant in the free planet designs investigated during this study.

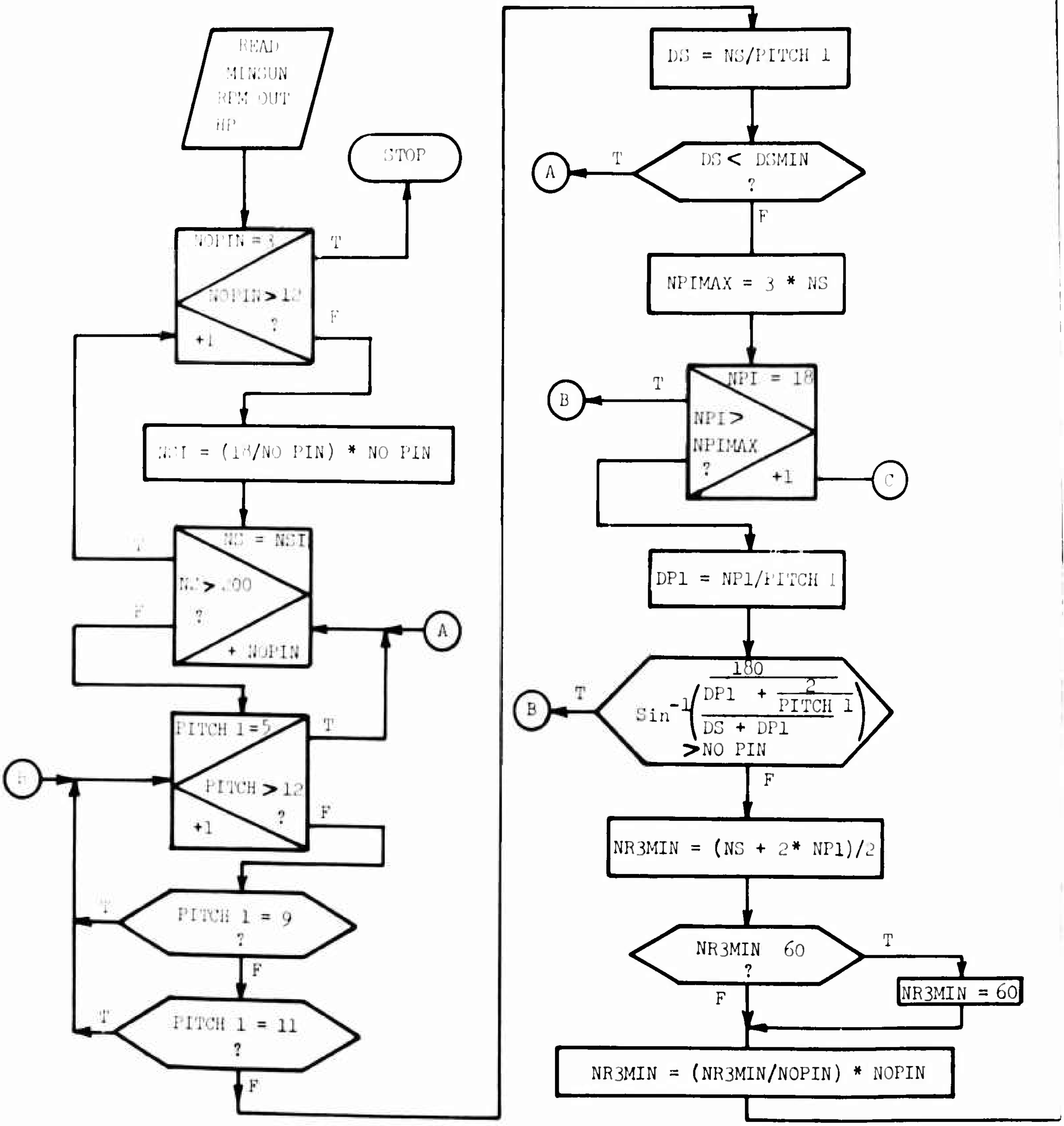
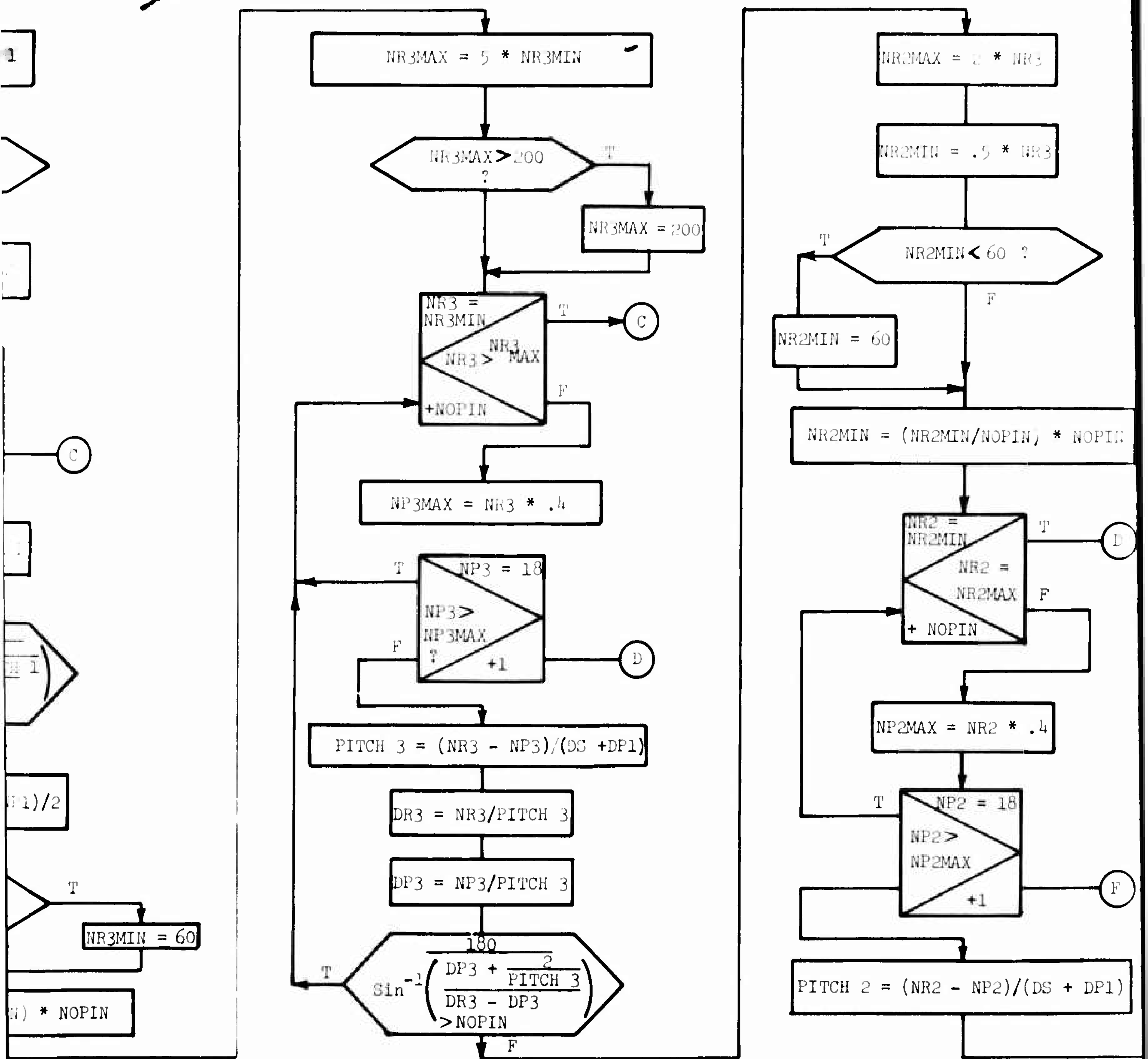
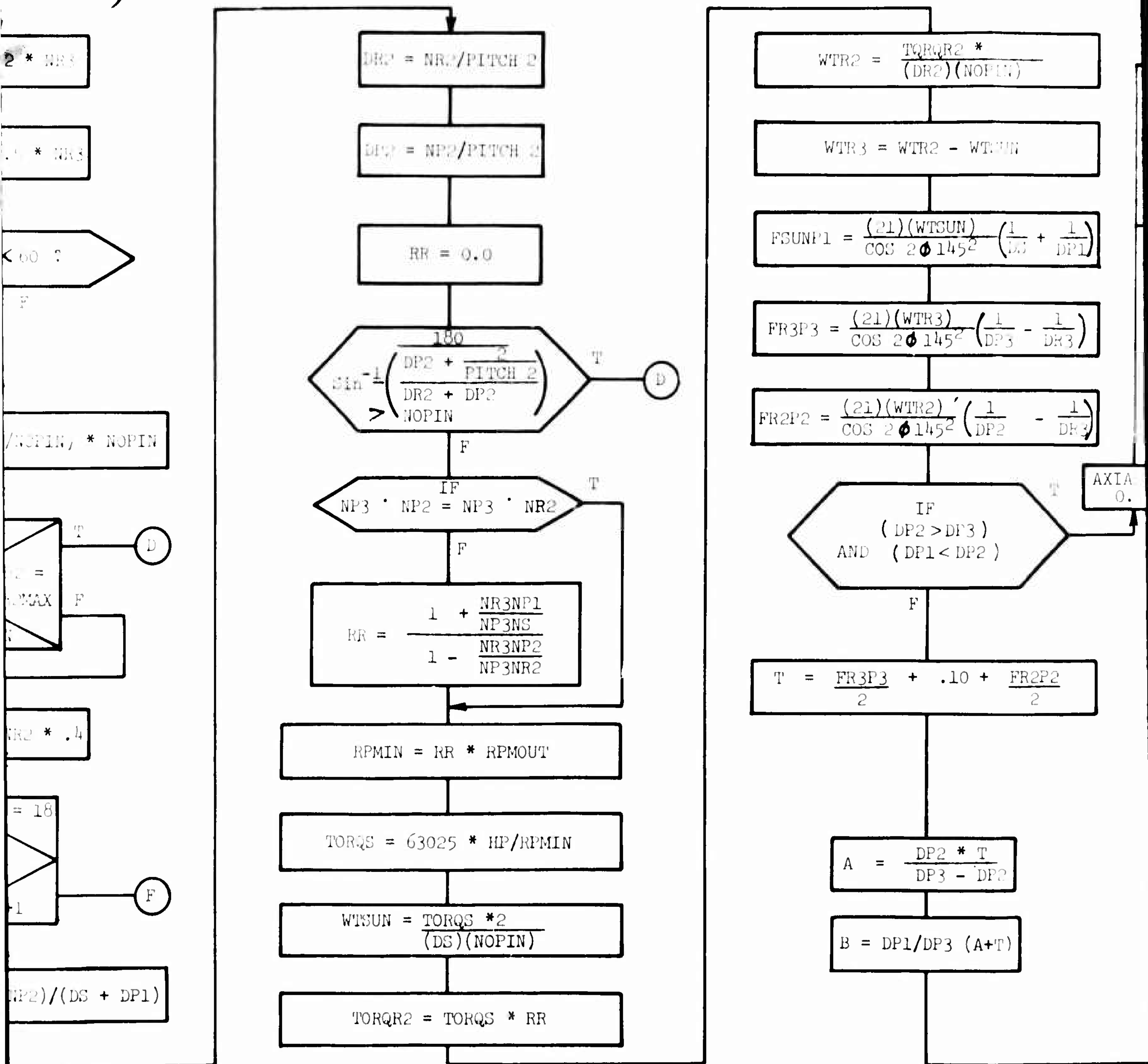


Figure 22. Simplified Program Flow Chart

2


NR3MAX = 5 * NR3MIN
NR3MAX > 200 ?
T
NR3MAX = 200
NR3 = NR3MIN
NR3 > NR3 MAX
+NOPIN
T
C
F
NP3MAX = NR3 * .4
NP3 = 18
NP3 > NP3MAX ?
+1
T
F
D
PITCH 3 = (NR3 - NP3)/(DS +DP1)
DR3 = NR3/PITCH 3
DP3 = NP3/PITCH 3
180
Sin
DP3 +
2
PITCH 3
DR3 - DP3
> NOPIN
T
F
NR2MIN < 60 ?
T
F
NR2MIN = 60
NR2MIN = (NR2MIN/NOPIN) * NOPIN
NR2 = NR2MIN
NR2 = NR2MAX
+ NOPIN
T
F
D
NP2MAX = NR2 * .4
NP2 = 18
NP2 > NP2MAX
+1
T
F
PITCH 2 = (NR2 - NP2)/(DS + DP1)
C
NR3MIN = 60
T
* NOPIN

3


DR2 = NR2/PITCH 2
DP2 = NP2/PITCH 2
RR = 0.0
Sin-1 180 / DP2 + 2/PITCH 2 / DR2 + DP2 / NOPIN
T
D
F
IF NP3 · NP2 = NP3 · NR2
T
F
RR = 1 + NR3NP1/NP3NS / 1 - NR3NP2/NP3NR2
RPMIN = RR * RPMOUT
TORQS = 63025 * HP/RPMIN
WTSUN = TORQS *2 / (DS)(NOPIN)
TORQR2 = TORQS * RR
WTR2 = TORQR2 * / (DR2)(NOPIN)
WTR3 = WTR2 - WTSUN
FSUNP1 = (21)(WTSUN) / COS 2φ 14.5²
FR3P3 = (21)(WTR3) / COS 2φ 14.5²
FR2P2 = (21)(WTR2) / COS 2φ 14.5²
IF (DP2 > DP3) AND (DP1 < DP2)
T
F
AXIA
0.
T = FR3P3/2 + .10 + FR2P2/2
A = DP2 * T / DP3 - DP2
B = DP1/DP3 (A+T)
2 * NR3
* NR3
< 60 ?
F
* NOPIN
T
D
=
MAX
F
R2 * .4
= 18
F
1
P2)/(DS + DP1)

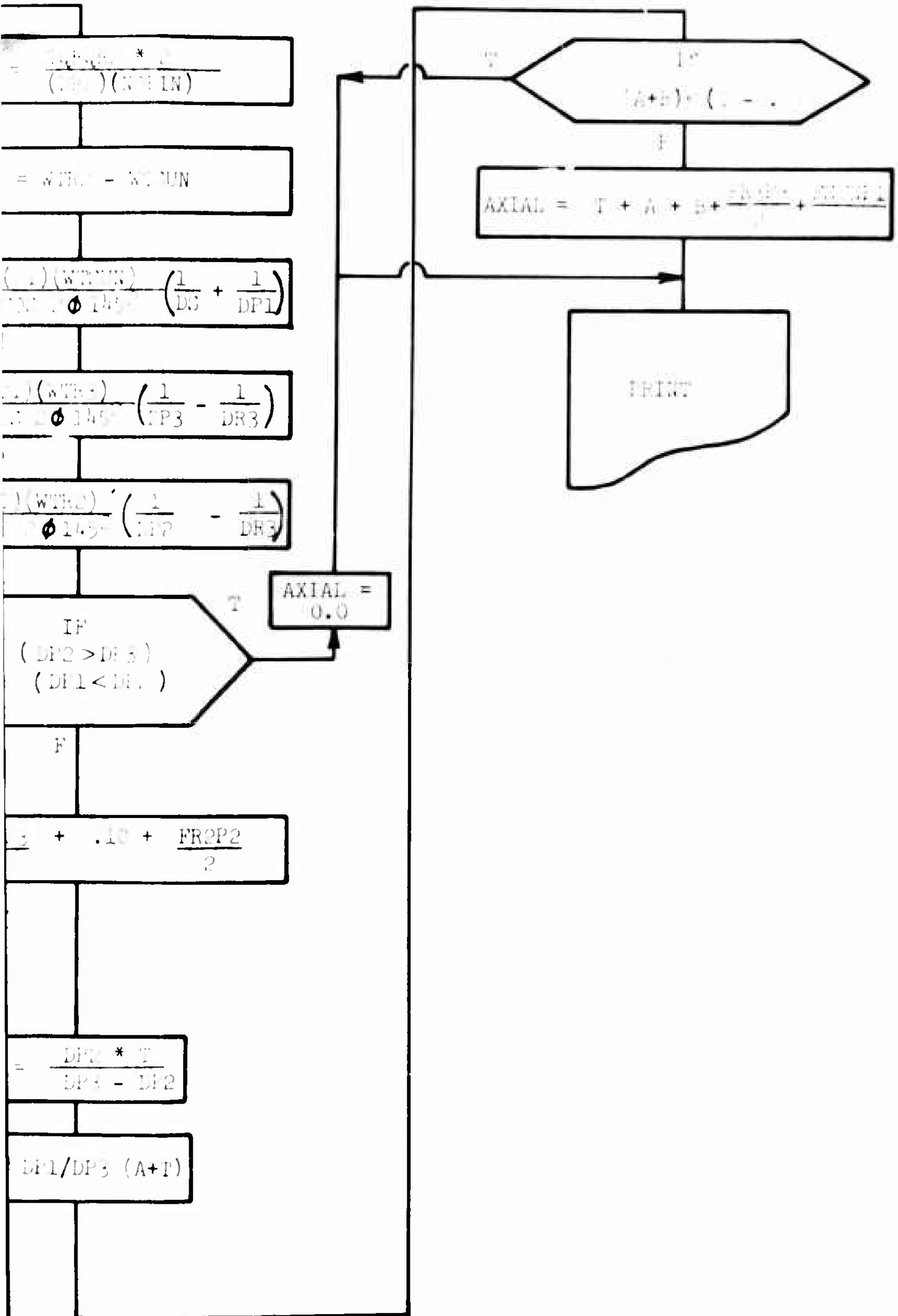

T
IF
F
AXIAL = T + A + B +
PRINT
AXIAL =
0.0
T
IF
F
FR2P2
2
(A+B)

TABLE 5. COMPUTER DESIGN SELECTION CRITERIA

ITEM		LIMITS LOW	LIMITS HIGH
Diametral Pitch*		5, ≠9 ≠11	12
Number of Pinions		5	12
Number of Teeth Sun Gear	S1	18	200
Diameter Sun Gear			$>$ 6.0
Number of Teeth Pinion	P1	18	
Diameter Pinion	P1		
Number of Teeth Pinion	P2	18	
Diameter Pinion	P2	$< D_{P3}$ & $D_{P1} > D_{P2}$	
Number of Teeth Ring Gear Output	R2	60	200
Diameter Ring Gear Output	R2		27.0
Number of Teeth Pinion	P3	18	
Diameter Pinion	P3		
Number of Teeth Ring Gear Reaction	R3	60	200
Diameter Ring Gear Reaction	R3		27.0
Reduction Ratio		+10.0	+40.0
Face Width of P2		$.6\ D_{P2}$	
Axial Length		$<$ 25.0 and 3 D_{R3}	
Hunting Teeth Criteria			

$$\frac{N_{S1}}{N_{P1}} = \text{Whole Number} + \text{Irreducible Fraction}$$

$$\frac{N_{R2}}{N_{P2}} = \text{Whole Number} + \text{Irreducible Fraction}$$

$$\frac{N_{R3}}{N_{P3}} = \text{Whole Number} + \text{Irreducible Fraction}$$

Equal Pinion Spacing Criteria for Each Mesh

$$\frac{N_{S1} + N_{R1}}{\text{No of Pinion}} = \text{Whole Numbers}$$

* The diametral pitch for all meshes was selected to be within ±.001 of a whole integer diametral pitch.

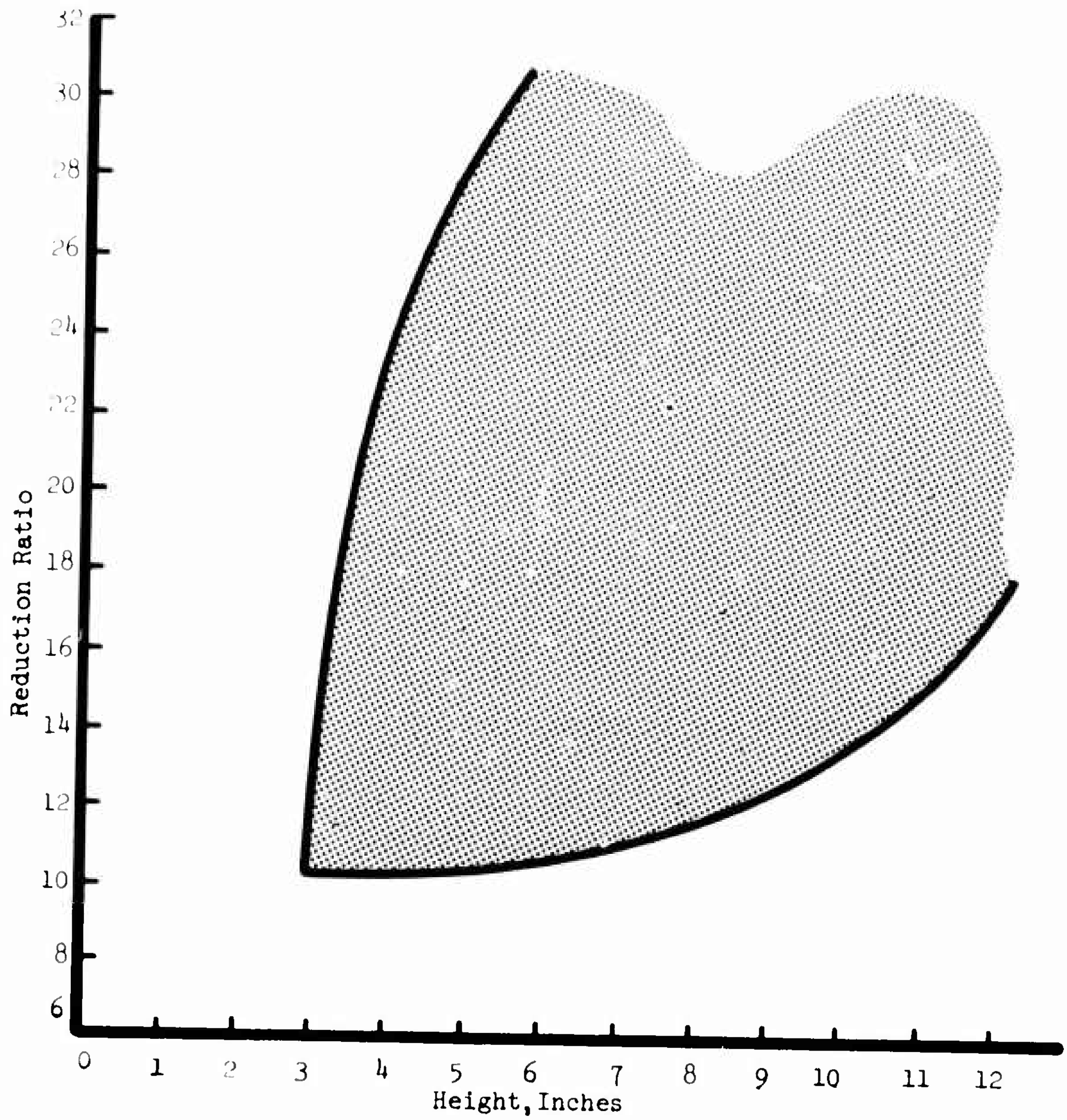


Figure 23. Map of Range of Free Planet Solutions

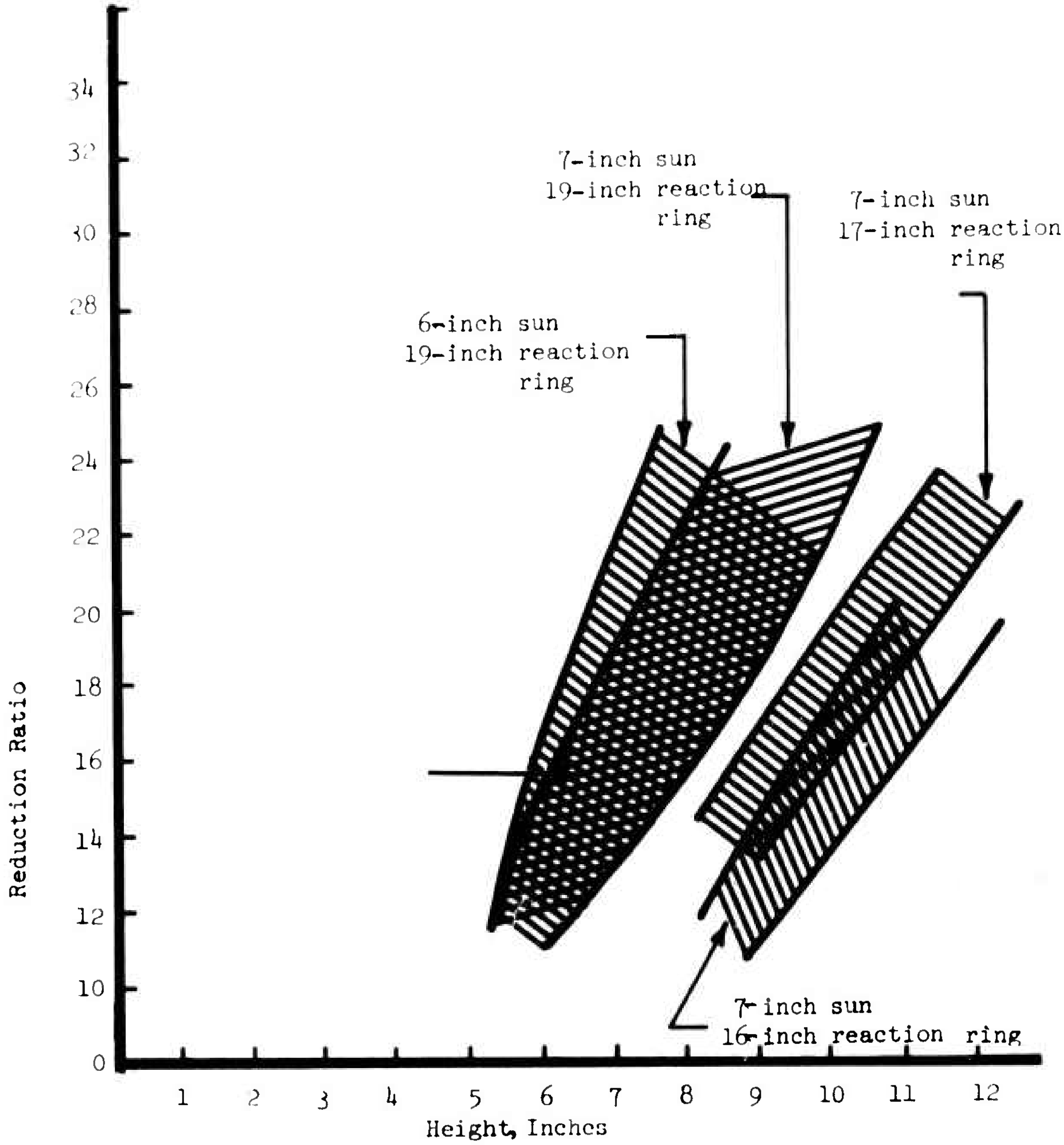


Figure 24. Reduction Ratio Versus Height for Given Sun and Ring Gears

PRELIMINARY LAYOUT

Conventional Two-Stage Planetary Transmission

A conventional two-stage planetary transmission was designed in order to compare a free planet transmission with conventional drive train technology. The conventional two-stage planetary drive consists of an input spur mesh bevel gear mesh, and two planetary reduction stages, one of which drives the main rotor. Figure 25 is a preliminary layout of the conventional planetary transmission. Power is transmitted to an input spur gear from each engine, which in turn drives a spur combining gear. This spur combining gear is located on the bevel pinion shaft and drives the bevel output gear, which is concentric with the main rotor shaft. The output of the bevel gear drives the sun gear of the first-stage planetary gear set. The planetary reduction stage has an input sun gear, stationary (bolted to housing) ring gear, and output cage. The output cage of the first-stage planetary drives the sun gear of the second-stage planetary. The cage output of the second-stage planetary is splined to the main rotor shaft and transmits power to the main rotor. The main rotor shaft is supported at the top of the main gearbox by a cylindrical roller bearing and at the bottom by a tandem-mounted, split inner race ball bearing set. Rotor loads are reacted through the main rotor shaft bearings to a cast magnesium housing bolted to the airframe. Figure 26 is a schematic of the drive train, showing the rotational speeds. The weight, which reflects a parametric weight for overrunning clutch and lubrication system, of the conventional planetary, is 737 pounds.

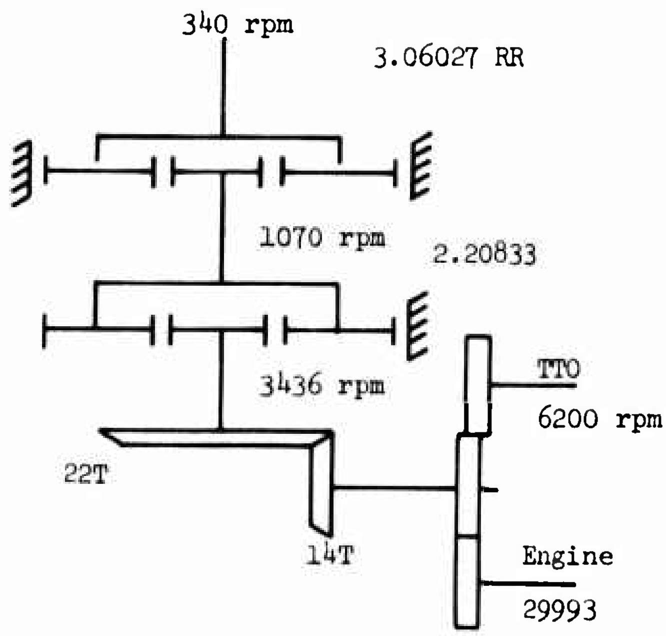


Figure 26. Schematic of Conventional Planetary Transmission

Figure 25. Preliminary Layout of Conventional Planetary Transmission

Free Planet Transmission

The free planet transmission consists of an input spur mesh from each engine, a bevel gear mesh, and the free planet reduction unit that drives the main rotor. Figure 27 is a schematic of the drive train, and Figure 28 is a preliminary layout of the free planet transmission.

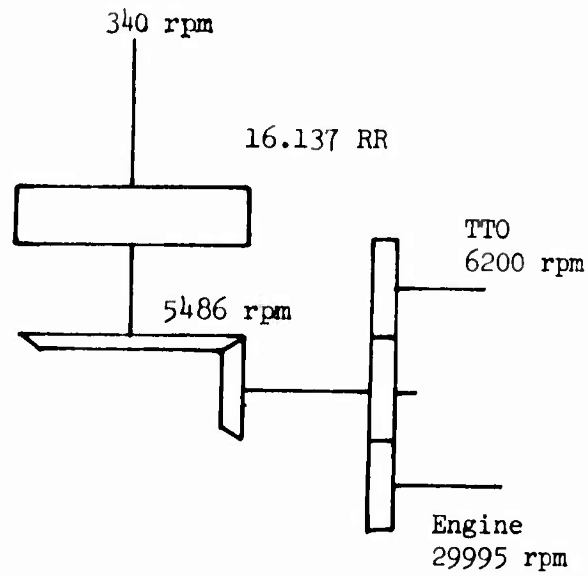


Figure 27. Preliminary Free Planet Drive Train Schematic

Power is transmitted from each of the two aft-mounted engines through an input spur gear to a spur combining gear. The spur combining gear is located on the bevel pinion shaft and drives the bevel output gear, which is concentric with the main rotor shaft. The spline output of the bevel gear transmits power to the sun gear of the free planet reduction unit. The free planet consists of an input sun gear, five planet pinion assemblies with three pinion gears for each shaft, three roller rings, and two ring gears. One ring gear provides the reaction torque. The other ring gear serves as the output member and transmits power to the main rotor. Table 6 supplies the free planet unit geometry.

The main rotor shaft, which is driven by the output ring gear, is supported at the top of the gearbox by a cylindrical roller bearing and at the bottom by a tandem-mounted, split inner race ball bearing set. Rotor loads are reacted through the main rotor shaft bearings to a cast magnesium gearbox housing. The weight for this preliminary free planet transmission includes a parametric weight for the overrunning clutch and lubrication system and is 705 pounds.

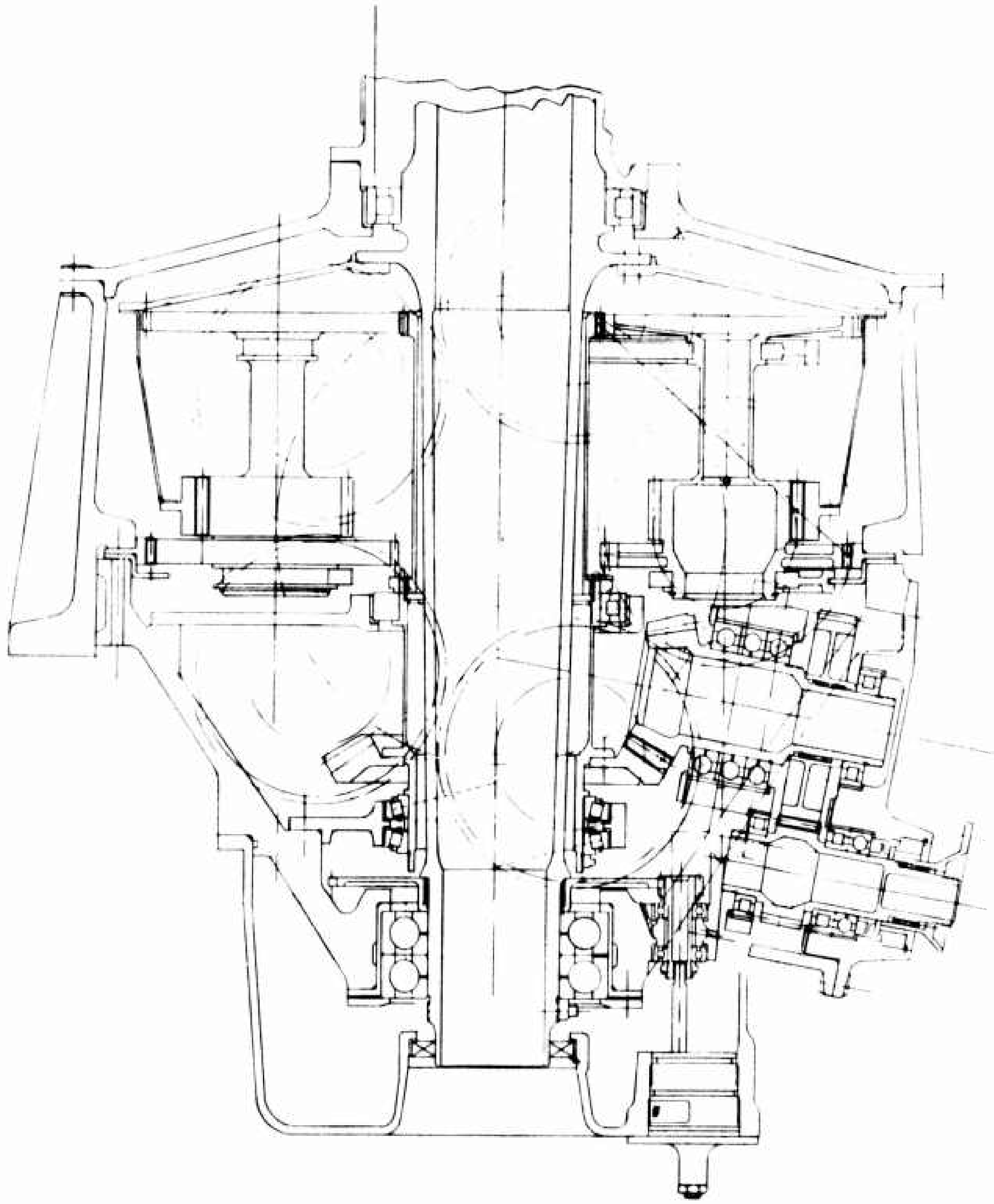


Figure 28. Preliminary Layout of Free Planet Transmission

TABLE 6. FREE PLANET UNIT GEAR TEETH GEOMETRY (PRELIMINARY SELECTION)					
ITEM	NUMBER OF TEETH	PITCH DIAMETER	DIAMETRAL PITCH	NUMBER OF PINIONS	REDUCTION RATIO
NS	50	5.0	10.0		
NP1	68	6.8			
NP2	27	3.84	7.034	5	16.137
NR2	110	15.64			
NP3	58	6.40	9.068		
NR3	165	18.20			

The weight of the free planet transmission is within 5% of the weight of a conventional planetary transmission. The computer model was then used with parametric curves to design a lower ratio free planet transmission with larger diameter ring gears.

FINAL FREE PLANET DESIGN

The final free planet transmission design is shown in Figure 29. The transmission is lower in height. It has a lower reduction ratio, larger sun gear diameter, and larger ring gear diameters. Figure 30 is a schematic showing the drive train speeds of the final design.

Table 7 supplies the gear geometry.

The gear tooth loads for an output speed of 340 rpm with 1,454 HP to the main rotor are

$$T_{out} = \frac{(63025)(1454)}{340} = 269520$$

$$T_{in} = \frac{269520}{10.851} = 24800$$

The tangential tooth load on the sun pinion mesh is

$$W_{ts} = \frac{2\ Tin}{d_s\ M}$$

$$= \frac{(2)(24800)}{(7.0)(5)}$$

$$W_{ts} = 1418$$

The tangential tooth load on the output ring gear mesh is

$$W_{t_{R_2}} = \frac{2\ Tout}{d_{R2}\ M}$$

$$= \frac{(2)(269520)}{(19.0)(5.0)}$$

$$W_{t_{R_2}} = 5674$$

The reaction ring gear torque is found from

$$W_{t_{R_3}} = W_{t_{R_2}} - W_{ts} = 4256$$

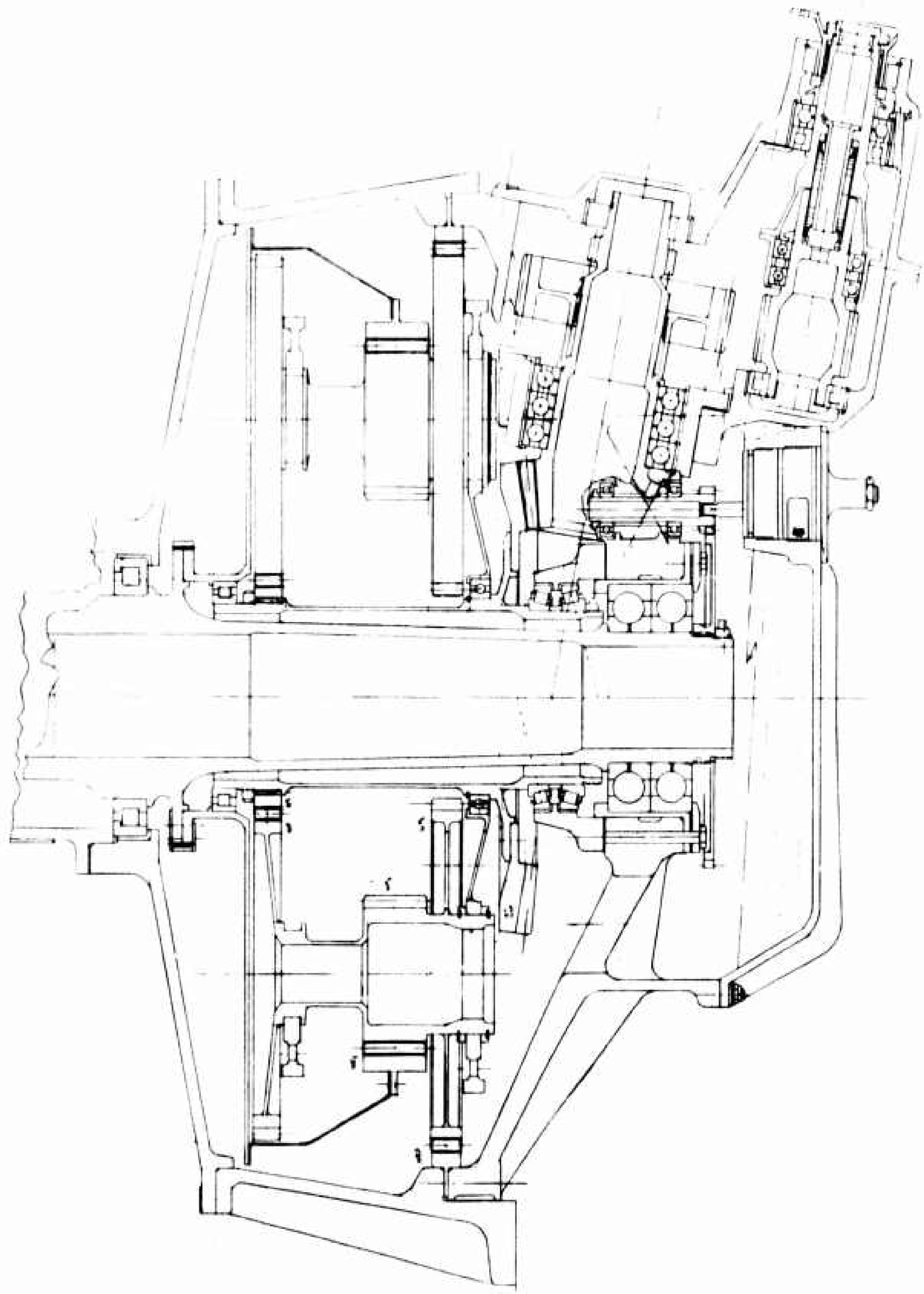


Figure 29. Layout of Final Free Planet Transmission

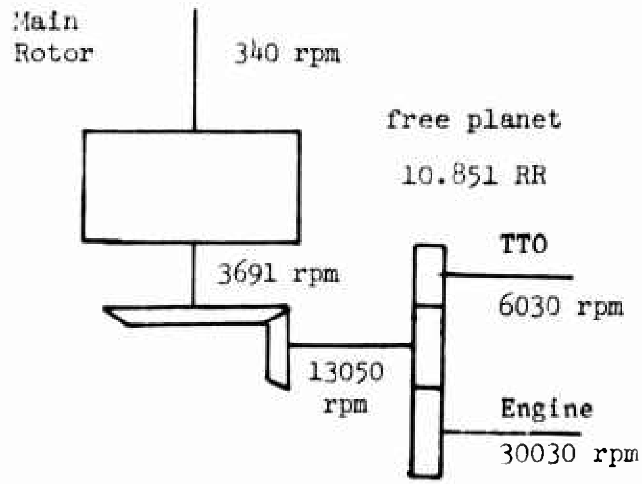


Figure 30. Schematic of Final Free Planet Transmission

TABLE 7. GEAR GEOMETRY FREE PLANET TRANSMISSION
REDUCTION RATIO 10.857

Member	Number of Teeth	Diametral Pitch	Number of Pinions	Pitch Diameter
Sun	35	5.0		7.0
P_1	39			7.8
P_2	21	5.0	5	4.2
R_2	95			19.0
P_3	41	5.0		8.20
R_3	115			23.0

The speed of the input sun gear is

$$N_{sun} = N_{out} \quad RR$$

$$N_{sun} = (340)(10.857)$$

$$N_{sun} = 3691 \quad \text{RPM}$$

The rotational speed of the pinion shaft about its own center is

$$N_{P_1P_2P_3} = \frac{1}{1 + \frac{P_1R_3}{P_3\ S}} \quad N_{sun}$$

$$N_{P_1P_2P_3} = \frac{1}{1 + \frac{(39)(115)}{(41)(35)}} \quad (3691)$$

$$N_{P_1P_2P_3} = 895 \quad \text{RPM}$$

Using these data, the gear tooth stresses and face width were calculated as outlined in the Gear Tooth Stress Analysis section. This information is summarized in Table 8.

TABLE 8. GEAR FACE WIDTH AND GEAR STRESSES

Member		Face Width	Compressive Stress	Bending Stress
Sun		.550	145,000	32,800
P_1		.545		32,050
P_2	Output	1.50	145,000	49,700
R_2	Ring Mesh	1.47		40,550
P_3	Reaction	.80	115,000	55,000
R_3	Ring Gear Mesh	.75		55,000

Weight Analysis

Comparison of the free planet transmission with a conventional two-stage planetary indicates that a 9% weight saving is achieved by the free planet design. The baseline conventional two-stage planetary design weighs 737 pounds, while the free planet transmission weighs 671 pounds. Table 9 contains a component weight breakdown for both transmissions.

The weights of the baseline main gearbox, main rotor shaft, and other drive system components were estimated from statistical weight trending equations based on Sikorsky's drive system design philosophy.

TABLE 9. WEIGHT COMPARISON OF FREE PLANET WITH CONVENTIONAL TWO-STAGE PLANETARY TRANSMISSION

Component	Free Planet Weight, Lb	Conventional Two-Stage Planetary Weight, Lb
Main Rotor Shaft	62.0	83.0
Planet Assembly	144.0	159.0
Gears	41.0	58.0
Tail Takeoff and Accessories	37.0	37.0
Bearings	37.0	50.0
Freewheel Units	10.0	10.0
Housings and Sump	200.0	195.0
Lubrication System	98.0	103.0
Seals, Spacers and Retainers	22.0	22.0
Supports	11.0	11.0
Miscellaneous	9.0	9.0
	671.0	737.0

Cost Analysis

The cost of the free planet transmission is 16.5% less than that of a conventional two-stage planetary transmission in quantities of 500 or more. Table 10 gives the estimated manufacturing cost for transmissions built in quantities of 1, 50, 100, and 500 units. Table 11 is a breakdown of cost by components for a prototype free planet transmission and conventional planetary transmission.

TABLE 10. COST COMPARISON OF FREE PLANET TRANSMISSION WITH TWO-STAGE CONVENTIONAL PLANETARY TRANSMISSION

Quantity (units)	Free Planet (dollars)	Conventional Planetary (dollars)
1	157,500	184,200
50	84,600	101,800
100	76,500	91,500
500	59,700	71,600
Prototype Tooling	254,800	252,000
Production Tooling (50 units or more)	645,000	631,000

TABLE 11. COST BREAKDOWN FOR PROTOTYPE FREE PLANET AND CONVENTIONAL PLANETARY TRANSMISSION

ITEM	FREE PLANET (dollars)		CONVENTIONAL PLANETARY (dollars)	
Raw Material		49,700		49,000
Castings	42,000		39,200	
Shaft Forgings	4,900		5,600	
Gear Forgings	2,800		4,200	
Bearing, Seals, "O" Rings		14,000		16,800
Spacers, Studs, Misc. Part		9,800		11,200
Scrap		4,200		5,600
Machining @20 $/hr.		70,000		89,600
Assembly @20 $/hr.		9,800		11,200
Total Prototype Cost		157,500		183,400
Tooling for Prototype		254,800		252,000

Survivability/Vulnerability Analysis

Assessment of survivability characteristics indicates that the free planet transmission offers improved operation after loss of lubrication compared with the conventional transmission. The free planet transmission is more vulnerable to 23mm AP threats than the conventional transmission.

For operation after loss of normal oil supply, the free planet transmission design offers several improvements. The most important of these is elimination of planet gear bearings, which are the major source of failure after loss of lubrication. The gears in the free planet unit are also more tolerant of oil loss. Fewer parts are used, so there are fewer sources of heat. The system is more efficient, so less heat is generated. The transmission density is low, so more cooling air is available.

The higher vulnerability of the free planet to 23mm threats compared with the conventional transmission reflects the vulnerability of roller rings. Protection of these rings would be difficult. To stop all threats would require massive rings which would be very heavy. One way to prevent this problem is to permit the projectile to perforate the ring without stopping the projectile or absorbing the energy.

Efficiency

The efficiency of the free planet transmission was compared with that of the conventional two-stage planetary drive. Overall transmission efficiency of the free planet transmission was 97.3%. Overall efficiency of the conventional planetary drive was 96.8%. This improvement in efficiency for the free planet transmission translates into 7.5 HP.

The efficiency of the free planet system was calculated by the equivalent system method. The term "equivalent" refers to the fact the tooth mesh losses in the two systems are the same. In the equivalent system approach, an artificial rotation is imposed on the complete planetary gear train to effectively stop the planet carrier. Relative motions of all members of the planetary train are unchanged. However, the planet gears are idlers in the equivalent system, and the entire train can be considered a conventional gear train with fixed axes of rotation. The only change is in the pitch-line velocities of the gears. They are now equal to the velocities of engagement of the gears in the planetary train.

By this approach, efficiency, speed, and power flow relationships can be determined for the free planet system.

The planetary gear train and the equivalent fixed-axes gear train are shown in Figure 31.

The reduction ratio of planetary gear trains is calculated as follows:

$$R = \frac{1 + \frac{N_B}{N_A}\,\frac{N_E}{N_D}}{1 - \frac{N_C}{N_F}\,\frac{N_E}{N_D}}$$

$$= \frac{1 + \frac{39}{35}\,\frac{115}{41}}{1 - \frac{21}{95}\,\frac{115}{41}} = \frac{76}{7} = 10.8571$$

The cage speed ratio is

$$\frac{W_c}{W_i} = \frac{1}{1 + \frac{N_B}{N_A}\,\frac{N_E}{N_D}}$$

$$= \frac{1}{1 + \frac{39}{35}\,\frac{115}{41}} = \frac{287}{1184} = .2424$$

$N_A = 35$

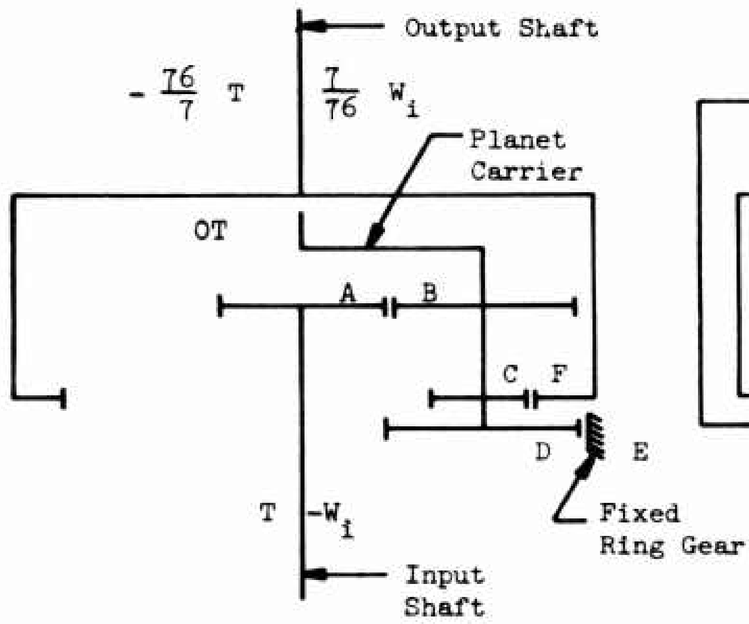
$N_B = 39$

$N_C = 21$

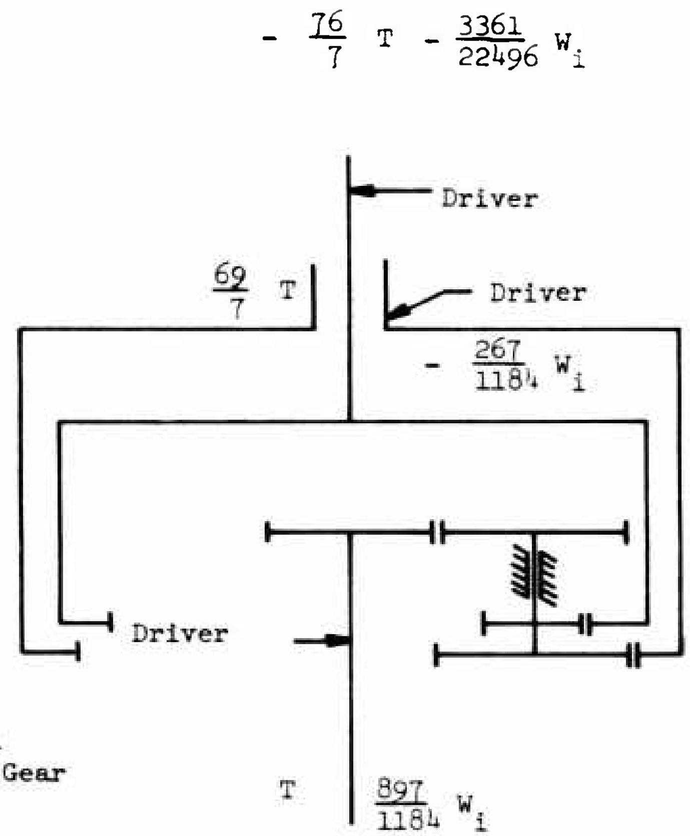
$N_D = 41$

$N_E = 115$

$N_F = 95$



Planetary Gear Train



Equivalent Gear Train

Figure 31. Free Planet Actual and Equivalent Fixed System Schematic

In Figure 31, the external torques acting on the shafts of the equivalent system are the same as those acting on the same shafts in the planetary system. The speeds in the equivalent system have been reduced by the carrier speed of the actual system. The product of the torque and the speed is a measure of the power transmitted. A positive value of this product indicates that the shaft is a driver. A negative value indicates that the shaft is driven. Thus, the equivalent gear train has two driving members and one driven member, and the power flow is from drivers to driven.

To determine system losses, the power developed at the two driving members must first be established. In the actual system, output power is

$$P_o = TW_i \eta_o$$

where η_o is the overall planetary efficiency. In the equivalent system, the output shaft is a driver and its power input is

$$\frac{76}{7} \left\{ \frac{3381}{22496} \right\} TW_i \eta_o = \frac{483}{296} TW_i \eta_o$$

The power input of the other driver is

$$\frac{897}{1184} TW_i$$

The power losses in the equivalent system can now be calculated:

$$\text{Losses} = \frac{483}{296} TW_i \eta_o \left(1 - \epsilon_{FC} \epsilon_{DE}\right) + \frac{897}{1184} TW_i \left(1 - \epsilon_{AB} \epsilon_{DE}\right)$$

where the E's are the appropriate fixed center gear mesh efficiencies.

The overall efficiency for the planetary gear then is

$$\eta_o = \frac{TW_i - \text{Losses}}{TW_i}$$

where the losses are the same as in the equivalent system.

$$\eta_o = \frac{TW_i - \frac{483}{296} TW_i \eta_o \left(1 - \epsilon_{FC} \epsilon_{DE}\right) - \frac{897}{1184} TW_i \left(1 - \epsilon_{AB} \epsilon_{DE}\right)}{TW_i}$$

Simplifying and solving for η_o,

$$\eta_o = \frac{1 - \frac{897}{1184} \left(1 - \epsilon_{AB} \epsilon_{DE}\right)}{1 + \frac{483}{296} \left(1 - \epsilon_{FC} \epsilon_{DE}\right)}$$

The fixed center gear efficiencies are calculated from

$$\varepsilon = 1 - \left\{ \frac{1 \pm \frac{M_1}{M_2}}{\beta_a + \beta_r} \right\} \left\{ \beta a^2 + \beta r^2 \right\} \frac{f}{2}$$

where $\frac{M_1}{M_2}$ = speed ratio

β_a, β_r = arc of approach and recess

f = average coefficient of friction

For mesh A - B shown in Figure 31:

$$\varepsilon_{AB} = 1 - \left\{ \frac{1 + \frac{N_A}{N_B}}{\beta_a + \beta r} \right\} \left\{ \beta a^2 + \beta r^2 \right\} \frac{f}{2}$$

$$\beta_a = \frac{\sqrt{R_{OB}^2 - R_{bB}^2} - R_B \sin \Phi}{R_{bA}}$$

$$= \frac{\sqrt{8.2^2 - 7.2063^2} - 7.8 \sin 22.5}{6.4672}$$

$$= .14346$$

$$\beta r = \frac{\sqrt{R_{OA}^2 - R_{bA}^2} - RA \sin \Phi}{R_{bA}}$$

$$= \frac{\sqrt{7.4^2 - 6.4672^2} - 7.0 \sin 22.5}{6.4672}$$

$$= .14191$$

$$V = \frac{D_A \pi N_A}{12}$$

$$= \frac{7\pi}{12}\left(\frac{897}{1184}\,\frac{76}{7}\right) 340 = 5125.1 \text{ fpm}$$

$$V_s = V \cos\phi \left\{1 + \frac{N_A}{N_B}\right\}. \frac{\beta_a + \beta_r}{4}$$

$$= (5125.1) \cos 22.5 \left(1 + \frac{35}{39}\right) .07134$$

$$= 641 \text{ fpm}$$

$$f = \frac{2}{3}\left\{\frac{.050}{\frac{641}{8}} + .002\sqrt{641}\right\}$$

$$= .03376$$

$$\varepsilon_{AB} = 1 - \left\{\frac{1 + \frac{35}{39}}{.28537}\right\}(.04072)\left(\frac{.03376}{2}\right)$$

$$= .9954$$

TABLE 12. EFFICIENCY DATA FOR FREE PLANET TRANSMISSION

MESH	B_a	B_R	V fpm	's	f	E
A - B	.14346	.14191	5121	641	.03376	.9954
C - F	.04918	.06384	2759	254	.02124	.9978
D - E	.14593	.12306	5388	215	.01957	.9991

The overall free planet efficiency is

$$= \frac{1 - \frac{897}{1184}\left(1 - (.9954)(.9991)\right)}{1 + \frac{483}{296}\left(1 - (.0078)(.9991)\right)}$$

$$= .9908$$

The analytical technique for calculating efficiency was compared with test data from the Curtiss-Wright 500 HP FP501 test unit. The analytical and test data agreed within .1%, which is well within the error of experimental measurement. The 19.2425:1 reduction ratio of the FP501 test unit had a measured efficiency of 98.8% of full speed and rated torque.

The calculated overall free planet efficiency for the final design configuration is 99.08%. The conventional two-stage planetary unit has an efficiency of 98.5%.

Total transmission losses were estimated from a knowledge of the type of gear mesh, design horsepower, and power transmitted by each gear mesh. Experience has demonstrated that tooth mesh and bearing losses can be estimated conservatively as 1/2% per mesh. A further 3/4% of total power transmitted must then be added to account for churning losses in the entire transmission.

The losses in the free planet and conventional planetary are presented in Table 13.

TABLE 13. LOSS, SOURCES AND EFFICIENCY OF FREE PLANET AND CONVENTIONAL TWO-STAGE PLANETARY DRIVES

	FREE PLANET		TWO-STAGE PLANETARY	
	LOSS PERCENT	LOSS HP	LOSS PERCENT	LOSS HP
MAIN ROTOR DRIVE-1450 HP				
EPICYCLIC	.92	13.34	1.5	21.75
BEVEL	.50	7.25	.50	7.25
SPUR	.50	7.25	.50	7.25
CHURNING	.75	10.87	.75	10.87
TAIL ROTOR DRIVE-110 HP				
BEVEL	.50	.55	.50	.55
SPUR	.50	.55	.50	.55
CHURNING	.75	.83	.75	.83
ACCESSORY DRIVE-30 HP				
BEVEL	.50	.15	.50	.15
SPUR	1.00	.60	1.00	.60
CHURNING	.75	.23	.75	.23
TOTAL LOSSES		41.62		50.03
EFFICIENCY		97.3		96.8

Reliability Analysis

The reliability of the free planet transmission was compared with that of a conventional two-stage planetary transmission. The analysis indicated a 2-to-1 improvement in reliability of the free planet over the conventional two-stage planetary.

Background

The reliability data used covered Sikorsky main gearboxes with a cumulative total of over 300,000 flight hours. The design criteria for gear tooth stresses and design bearing lives were taken as identical in the free planet and the conventional planetary. Failure rates for the component elements (sun gears, planet pinions, ring gears, bearings) were taken as identical for the two systems.

Both the free planet and conventional two-stage planetary were considered to have the same failure modes as those experienced by production planetary designs. The measure of reliability is the removal rate caused by a validated failure before a gearbox reaches its scheduled removal time for overhaul, so the removal rate does not include scheduled removals.

The failure modes experienced in the sample of operational main transmissions include gear tooth fracture, spalling, scoring, and wear, in addition to planet pinion bearing failures. Of these failure modes, only gear tooth fracture and planet pinion bearing failures result in gearbox removals. This permits the use of gear tooth bending stress and bearing failures in the reliability analysis.

The failure rates based on flight test data are shown in Table 14.

TABLE 14. FAILURE RATE FOR GEARS AND BEARINGS BASED ON OPERATIONAL DATA

MEMBER	FAILURE RATE (failures/cycle)
Sun Gear	31.5×10^{-12}
Planet Pinion	177.5×10^{-12}
Ring Gear	Negligible
Pinion Bearings	16.7×10^{-12}

These failure rates are expressed in failures per cycle to take into account the number of mesh points of the gears as well as the frequency of loading.

Analysis

The gear tooth stress levels and resulting failure rates are not precisely those of the free planet transmission or conventional planetary transmission. The rates have been adjusted to provide a valid comparison. This adjustment reflects the assumption that the ratio of actual failure rate to probability of failure, based on working stress level, is constant for each type of gear. Therefore, a constant can be defined for sun gears, planet pinions, and ring gears. In establishing this constant from past operational designs, the fatigue bending endurance strength of the gear teeth was taken as the endurance strength of the core material for a part with a ground surface.

The endurance limit was determined as follows:

Material		9310 CEVM Steel
Core Hardness		R_c 30-45
Endurance Limit E_m (Ground surface)		55,000 psi
Size Effect Factor (SEF) (based on geometry)		0.8
Mean Endurance Limit	=	$(SEF)(E_m)$
	=	(.8)(55,500)
	=	44,400 psi

This mean endurance limit of 44,000 psi is used in the design of gearboxes and represents a 50% probability of failure. Extensive test experience has established that the standard deviation of fatigue data for steel is 10% of the mean strength for components made of the same and of similar steel alloys. The probability of failure for any other stress level can now be determined easily from standard tables of the Gaussian distribution. These tables give probabilities as a function of the number of standard deviations.

The quantity $a\sigma$ is defined as

$$a\sigma = \frac{(\text{Mean Endurance Limit})-(\text{Working Stress})}{(\text{Mean Endurance Limit})\ (\text{Standard Deviation})}$$

Figure 32 represents the basic concepts of the analytical approach.

As an example of this analysis for the free planet transmission, consider the sun-pinion mesh. The sun gear failure rate from Table 14 is 31.5×10^{-12} failures per cycle, based on a maximum working stress level of 30,500 psi. Based on a normal statistical distribution for a mean endurance limit of 55,000 psi and a standard deviation of 10% of the mean stress, the probability of failure at this stress is .0009. Defining a constant, K, as the ratio of failure rate to the probability of failure gives

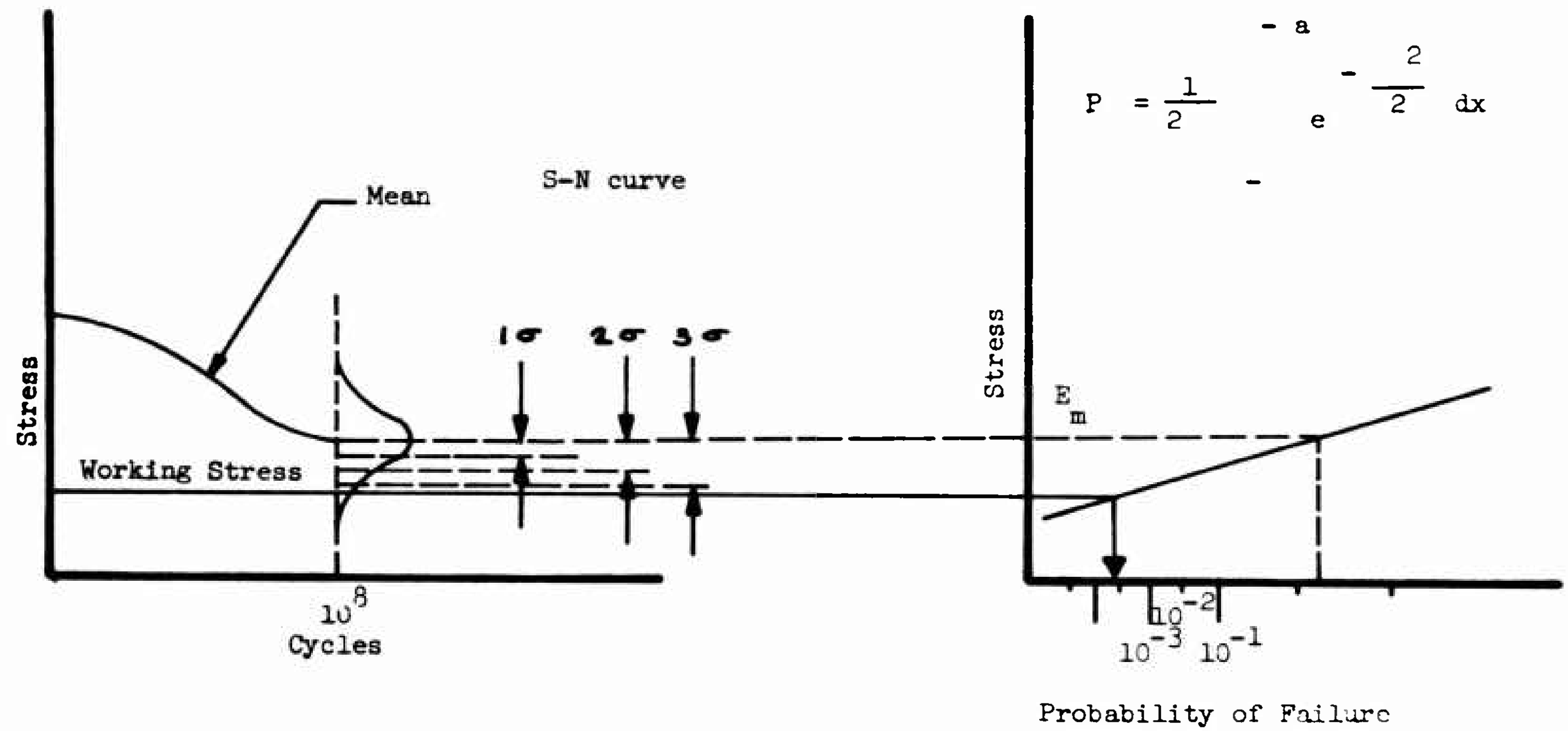


Figure 32 . Concept of Reliability Approach

$$K_s = \frac{\lambda_s}{P_s}$$

$$= \frac{31.5 \times 10^{-12}}{.0009}$$

$$= 3.5 \times 10^{-8}$$

Similarly, in the case of a planet pinion failure, the failure rate is based on a maximum stress of 34,800 psi (P_p = .0150), which leads to

$$K_p = 1.5 \times 10^{-8}$$

The total failure rates of the free planet transmission and the conventional two-stage planetary can be found by combining the individual component failure rates with the number of components.

For the free planet transmission,

$$\begin{aligned}
\lambda_{\text{Free Planet}} = \; & \lambda_{\text{sun}} + \left(m_{\text{sun/pinion}} \, \lambda_{\text{sun/pinion}}\right) \\
& + \left(m_{\text{pinion/ring}} \, \lambda_{\text{pinion/ring}}\right) \\
& + \left(m_{\text{output pinion/ring}} \, \lambda_{\text{output pinion/ring}}\right) \\
& + \left(m_{\text{upper roller}} \, \lambda_{\text{rollers}}\right) \\
& + \left(m_{\text{lower roller}} \, \lambda_{\text{rollers}}\right) \\
& + \lambda_{\text{ball bearing}} \\
& + \left(m_{\text{pinion shaft}} \, \lambda_{\text{shaft}}\right)
\end{aligned}$$

The term ball bearing is the failure rate of the bearing that supports the dead weight of the free planet. This failure rate was assumed to be negligible, since the bearing has a calculated life of over 12,000 hours.

For the conventional two-stage planetary design,

$$\begin{aligned}
\lambda_{\text{planetary}} = \; & \lambda_{\text{sun}} \cdot (2) + m_{\text{pinion}}\left(\lambda_{\text{sun/pinion}} + \lambda_{\text{pinion/ring}}\right) \\
& + m_{\text{thrust washers}} \, \lambda_{\text{thrust washers}} \\
& + m_{\text{plates}} \, \lambda_{\text{plates}} \\
& + m_{\text{bearings}} \, \lambda_{\text{bearings}} \\
& + \lambda_{\text{ring}} \; (2)
\end{aligned}$$

For the comparative analysis of a free planet with a conventional planetary, the endurance limit for current designs is 71,700 psi for ground surfaces. The "a" coefficients of σ to obtain the probabilities from the Gaussian distribution are calculated with this new endurance limit. All the historic data, as well as the actual comparative data, are summarized in Table 15.

The only potential problem that has not been addressed in this analysis is failure of rollers and journal of the free planet design. The rollers are expected to experience a pitting mode of failure, which would not result in gearbox removal.

The electron beam weld in the pinion shaft design is a fabrication technique recently introduced at Sikorsky. Electron beam welding was used for fabrication of rolling elements and gears on the Roller Gear Drive Development Program. In the roller gear drives built for that program, the design of the electron beam weld was the source of almost all problems. The welds were subsequently redesigned, and quality control procedure were improved to the point where after 79.5 hours of testing there were no weld failures.

The failure rates predicted for the free planet and conventional planetary designs are as follows:

Conventional Two-Stage Planetary	
First Stage	33.701×10^{-6}
Second Stage	11.837×10^{-6}
Total	45.538×10^{-6}
Free Planet	25.890×10^{-6}

The predicted mean time between failures (MTBF) is defined as the total flight hours on all parts, both satisfactory and failed, divided by the number of anticipated failures. MTBF is the reciprocal of the total failure rate. Therefore, for the two systems, the comparative predicted MTBF's are:

Conventional Two-Stage Planetary System	MTBF = 22,000 hours
Free Planet	MTBF = 38,600 hours

The preceding analysis has provided failure rates on components experiencing stress cycles at the greatest rate. The total failure rate is also affected by the number of components in each system. Since the free planet has far fewer components, and the criteria used were comparable for both designs, the results indicate that the MTBF of the free planet design is approximately twice that of the conventional planetary. The validity of these analyses can be determined only by testing in an environment representing the operational use as closely as possible. The real proof of the predicted reliability can be demonstrated only with a sample chosen from operational use.

TABLE 15. FAILURE RATES COMPARISON OF FREE PLANET AND A TWO-STAGE CONVENTION PLANETARY DRIVE UNIT

UNIT	FATIGUE BENDING STRESS	a	PROBABILITY OF FAILURE	$K10^8$	CYCLES PER HOUR $X\ 10^{-6}$	NUMBER PER HOUR $X10^6$	NUMBER OF COMPONENTS	TOTAL FAIL. RATE FAILURE/ HOUR $X10^6$
FREE PLANET								
Sun	32,800	5.43	10^{-7}	3.5	.167796	Negl.	1	-
Pinion(Sun)	32,050	5.53	10^{-7}	1.5	.150588	Negl.	5	-
Pinion(Output)	49,700	3.07	.0010704	1.5	.150588	2.418	5	12.090
Ring(Output)	40,550	4.34	.0000071	Negl.	.033288	Negl.	1	-
Pinion(Reaction)	50,000	3.03	.0012220	1.5	.150588	2.760	5	13.800
Ring(Reaction)	55,000	2.33	.0099031	Negl.	.053683	Negl.	1	-
Bearing	-	-		-		Negl.	1	-
TOTAL UNIT								25.89
CONVENTIONAL TWO-STAGE PLANETARY								
First Stage								
Sun	28,000	6.09	10^{-7}	3.5	.147000	Negl.	1	-
Pinion(Sun)	44,800	3.75	.00035259	1.5	.242340	1.282	7	8.974
Pinion(Ring)	44,200	3.84	.00025057	1.5	.242340	0.911	7	6.377
Ring	44,000	3.86	.0000567	Negl.	.066420	Negl.	1	-
Bearings	-			-	.242340	4.05	7	28.350
Second Stage								
Sun	30,000	5.82	10^{-7}	3.5	.046020	Negl.	1	-
Pinion(Sun)	44,900	3.028	.00036605	1.5	.073320	0.403	7	2.821
Pinion(Ring)	44,200	3.84	.0000615	1.5	.073320	0.068	7	0.476
Ring	52,000	2.75	.0029798	Negl.	.020400	Negl.	1	-
Bearings	-			-	.073320	1.22	7	8.540
TOTAL UNIT								55.50

Helicopter Design Modeling -- Analysis

The Sikorsky HDM (Helicopter Design Model) was used to evaluate the effect of reductions in transmission weight and cost on the MUT. HDM is a rapid, efficient tool for design iteration and evaluation of baseline helicopters and advanced concept helicopters.

Background

Preliminary design of an aircraft is an iterative procedure involving configuration, weights, and performance. An initial configuration is developed from such design constraints as payload, volume, number of crew, number of engines, limit on rotor size, and mission equipment.

HDM is a digital computer program that provides the designer with the following outputs: rotor geometry, component weight breakdown, mission analysis, engine and gearbox sizing, speed capability, and cost. These outputs provide the designer with the refinements needed for each design iteration. A closed solution is achieved when the configuration, performance, weights, mission requirements, and system design specifications are consistent. Thus, HDM plays an important part in closing the design loop and furnishes insight into design sensitivities at the preliminary level to a degree never previously realizable. Aside from the derivation of the design point aircraft, the extensive trade-off and optimization capability of HDM enables the designer to trend away from the baseline configuration.

The program is available on the UNIVAC 1110 facility at our corporate research laboratories in Hartford, Connecticut. The program has been the primary preliminary design tool for the following contracts and proposals:

- U.S. Army Advanced Antitorque Study
- U.S. Army HLH Proposal
- U.S. Army UTTAS Proposal
- NASA/Army Rotor Systems Research Aircraft Predesign Study
- U.S. Army Structural Armor Fuselage Study
- U.S. Army ABC Operational Configuration Study
- U.S. Navy VTOL Escort Study
- U.S. Army AAH Proposal

For the present study, HDM was modified to suit the design constraints for a medium-size utility helicopter (MUT) and to obtain the desired level of detail in weights equations, engine and gearbox sizing criteria, and aerodynamic performances.

HDM has four basic loops L0, L1, L2, L3, as shown in Figure 33. L0 is used to derive the gross weight needed to achieve the required payload. If gross weight is specified, payload is calculated. The calculations within L0 form the nucleus of the program. L1, L2, and L3 enable trending, for a single set of input data, of the three primary design contraints: blade loading (C_T/σ). Elements of the drive system may be sized on the

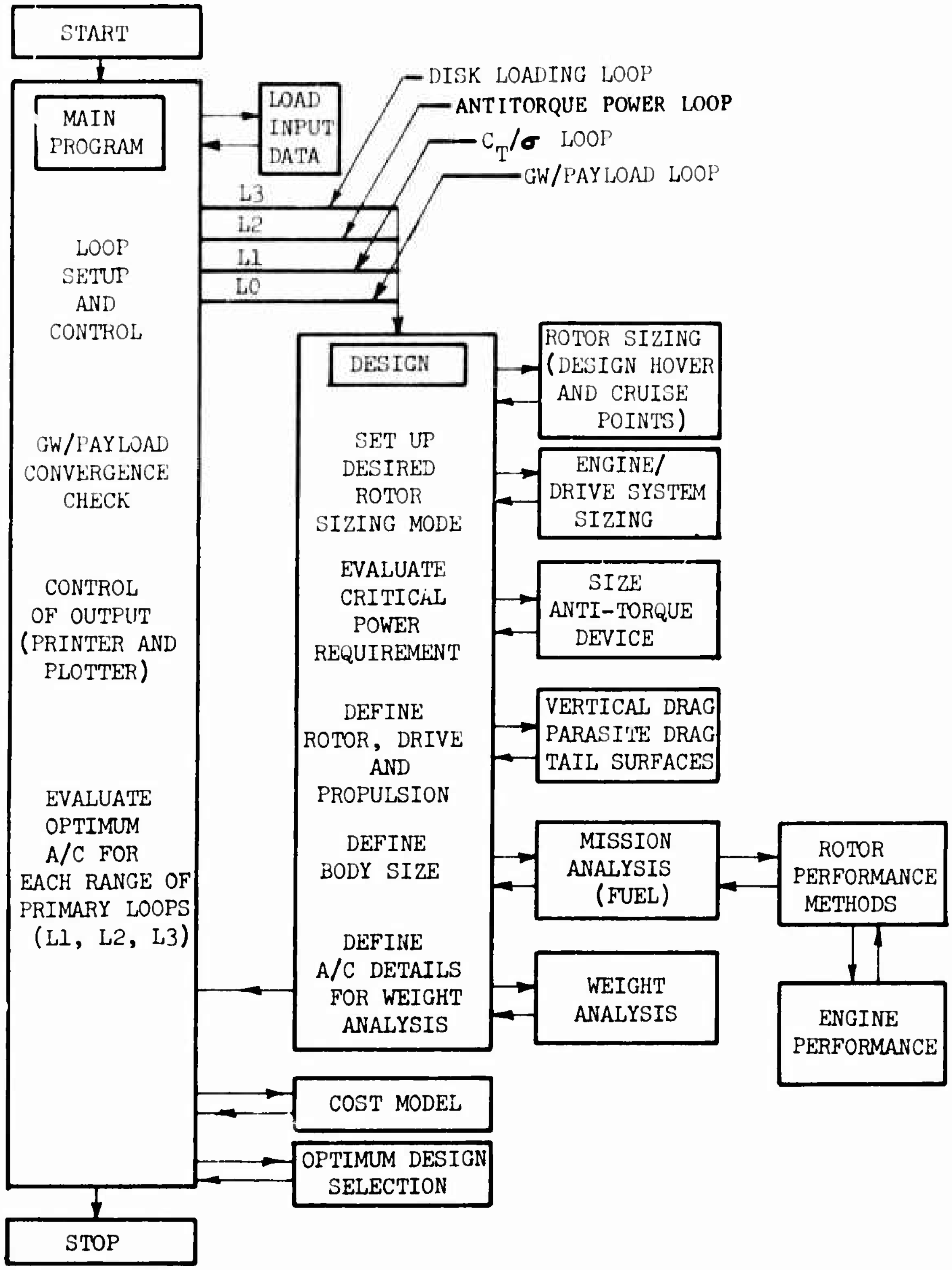


Figure 33. Helicopter Design Model Flow Chart

basis of a design performance requirement, such as percent over-rating above the design hover input power. Knowledge of rotor power and PCTPR defines total power required from the engines, thus enabling selection of engine type and size. If required rotor geometry (radius and chord) is specified, CTSIG and DL are calculated. If a particular tail rotor geometry is specified, PCTPR is calculated. CTSIG, DL, and PCTPR may be selected as single inputs or as a required range (initial, final, and incremental values) so that repeated passes are made around the appropriate loop (L1, L2 or L3) to create a matrix of design points. For each range of any of these three variables, the interpolated value needed to produce the aircraft selected, based on user perference for minimum weight, minimum cost, maximum productivity, etc. Thus, if ranges of values are desired for CTSIG, DL, and PCTPR, the program identifies the combination of values needed to optimize the helicopter design. The user may request printouts at various levels of definition and at varying frequency through the calculation. For example, he may request a complete detailed weight breakdown for every pass around L0, or a summary weight statement on completion of optimization.

Life-cycle cost of a military helicopter is a summation of the costs of development, production, ground support equipment, crew training, maintenance, spares, and fuel. The composition of each of these items depends on the particular project under study. Development and production costs for the baseline MUT helicopter were statistically trended and were, in general, a function of the component weights already calculated. Outputs from this subroutine are production cost, flyaway cost, and life-cycle cost. Cost modeling was limited to flyaway cost for the purpose of this study. Flyaway cost was based on production of 500 aircraft and is stated in 1974 dollars. Table 16 is the MUT baseline data sheet, Table 17 presents MUT baseline weights, and Table 18 presents MUT baseline costs.

For the MUT aircraft with the free planet transmission, the aircraft was resized for two different cases: (1) with the same payload as the baseline aircraft and (2) with the same gross weight as the baseline aircraft. In each of these cases, the dollars per pound and the weights of the total transmission system were changed to reflect the improvements with the free planet transmission. For the case with the same payload as the baseline aircraft, Table 19 is the summary data sheet for the resized MUT aircraft, Table 20 is the summary weight statement, and Table 21 is a life-cycle cost summary. Similarly, for the case with the same gross weight as the baseline aircraft, Table 22 is the summary data sheet for the resized MUT aircraft, Table 23 is the summary weight statement, and Table 24 is a life-cycle cost summary.

TABLE 16. MUT BASELINE DATA SHEET

DESIGN ATTRIBUTES

GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G.W. (LB)	9471.	RADIUS (FT)	20.50	RADIUS (FT)	4.40
PAYLOAD (LB)	960.	CHORD (FT)	1.322	CHORD (FT)	.535
WEIGHT EMPTY (LB)	6618.	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (LB)	1389.	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1178.	TIP SPEED (FPS)	730.0	TIP SPEED (FPPS)	700.0
HOVER + CLIMB HP	1261.	ASPECT RATIO	15.511	ASPECT RATIO	8.231
MAIN ROTOR DESIGN HP	1048.	CT/SIGMA	.0850	CT/SIGMA	.1089
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9239.4	TAIL ROTOR LIFT	231.9
M.R. DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7147
MAIN G.B. DESIGN HP	1564.	BLADE AREA (SQ.FT.)	108.4	BLADE AREA (SQ.FT.)	9.4

TABLE 17 . SUMMARY WEIGHT STATEMENT
BASELINE MUT

GROUP	WEIGHT		% GW
MAIN ROTOR GROUP		820.	8.65
WING GROUP		0.	.00
TAIL GROUP		152.	1.60
TAIL ROTOR/FAN	47.		.49
TAIL SURFACES	105.		1.11
BODY GROUP		1055.	11.14
ALIGHTING GEAR		380.	4.01
FLIGHT CONTROLS		638.	6.74
ENGINE SECTION		100.	1.06
PROPULSION GROUP		1907.	20.14
ENGINES	422.		4.46
AIR INDUCTION	40.		.42
EXHAUST SYSTEM	297.		3.13
LUBRICATING SYSTEM	0.		.00
FUEL SYSTEM	269.		2.84
ENGINE CONTROLS	25.		.26
STARTING SYSTEM	19.		.20
AUXILIARY PROPULSION PROPELLERS	0.		.00
DRIVE SYSTEM	835.		8.82
AUXILIARY POWER UNIT		0.	.00
INSTRUMENTS		135.	1.43
HYDRAULICS		0.	.00
ELECTRICAL GROUP		247.	2.61
AVIONICS		460.	4.86
ARMAMENT GROUP		53.	.56
FURNISHINGS		422.	4.46
AIR CONDITIONING AND ANTI-ICE		48.	.51
AUXILIARY GEAR		60.	.63
VIBRATION SUPPRESSION		76.	.80
TECHNOLOGY SAVINGS		0.	.00
CONTINGENCY		66.	.70
WEIGHT EMPTY		6618.	69.88
FIXED USEFUL LOAD		504.	5.32
PILOT	235.		
COPILOT	235.		
OIL-ENGINE	14.		
-TRAPPED	6.		
FUEL TRAPPED	14.		
MISSION EQUIPMENT	0.		
OTHER FUL.	0.		
PAYLOAD		960.	10.14
FUEL-USABLE		1389.	14.66
GROSS WEIGHT		9471.	

TABLE 18. LIFE-CYCLE COST SUMMARY - BASELINE

ITEM		DOLLARS
DEVELOPMENT COST PER AIRCRAFT		88959.
PROTOTYPE COST PER PRODUCTION AIRCRAFT		22291.
RECURRING PRODUCTION COST	529160.	
GFE AVIONICS	40000.	
ENGINE COST	89378.	
(FLYAWAY COST)	(658539.)	
INITIAL SPARES	206147.	
GROUND SUPPORT EQUIPMENT	39512.	
INITIAL TRAINING AND TRAVEL	52601.	
ACQUISITION COST		956800.
FLIGHT CREW	457200.	
FUEL + OIL	298324.	
REPLENISHMENT SPARES	893368.	
ORG + D/S + G/S MAINT	369534.	
DEPOT MAINTENANCE	322775.	
RECURRING TRAINING	274509.	
MAINTENANCE OF GSE	20769.	
OPERATING COST		2636479.
LIFE-CYCLE COST		3704529.
PRODUCTIVITY		.01088
FLEET LIFE CYCLE COST		1652264512.

TABLE 19. DATA SHEET
MUT AIRCRAFT RESIZED TO SAME PAYLOAD WITH
FREE PLANET TRANSMISSION WEIGHT AND COST

DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G.W. (LB)	9335.	RADIUS (FT)	20.35	RADIUS (FT)	4.37
PAYLOAD (LB)	960.	CHORD (FT)	1.312	CHORD (FT)	.531
WEIGHT EMPTY (LB)	6495.	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (LB)	1376.	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1162.	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	700.0
HOVER + CLIMB HP	1243.	ASPECT RATIO	15.511	ASPECT RATIO	8.229
MAIN ROTOR DESIGN HP	1033.	CT/SIGMA	.0850	CT/SIGMA	.1087
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9106.4	TAIL ROTOR LIFT	228.5
M.R. DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7146
MAIN G.B. DESIGN HP	1542.	BLADE AREA (SQ.FT)	106.8	BLADE AREA (SQ.FT)	9.3

SUMMARY WEIGHT STATEMENT
MUT AIRCRAFT RESIZED TO SAME PAYLOAD

GROUP	WEIGHT		% GW
MAIN ROTOR GROUP		808.	8.65
WING GROUP		0.	.00
TAIL GROUP		149.	1.60
TAIL ROTOR/FAN	46.		.49
TAIL SURFACES	104.		1.11
BODY GROUP		1049.	11.23
ALIGHTING GEAR		375.	4.02
FLIGHT CONTROLS		630.	6.75
ENGINE SECTION		100.	1.07
PROPULSION GROUP		1819.	19.49
ENGINES	418.		4.48
AIR INDUCTION	40.		.43
EXHAUST SYSTEM	295.		3.16
LUBRICATING SYSTEM	0.		.00
FUEL SYSTEM	267.		2.86
ENGINE CONTROLS	25.		.27
STARTING SYSTEM	19.		.20
AUXILIARY PROPULSION PROPELLER	0.		.00
DRIVE SYSTEM	756.		8.10
AUXILIARY POWER UNIT		0.	.00
INSTRUMENTS		135.	1.45
HYDRAULICS		0.	.00
ELECTRICAL GROUP		247.	2.65
AVIONICS		460.	4.93
ARMAMENT GROUP		53.	.57
FURNISHINGS		422.	4.52
AIR CONDITIONING AND ANTI-ICE		48.	.51
AUXILIARY GEAR		60.	.64
VIBRATION SUPPRESSION		75.	.80
TECHNOLOGY SAVINGS		0.	.00
CONTINGENCY		65.	.70
WEIGHT EMPTY		6495.	69.58
FIXED USEFUL LOAD		504.	5.40
PILOT	235.		
COPILOT	235.		
OIL-ENGINE	14.		
-TRAPPED	6.		
FUEL TRAPPED	14.		
MISSION EQUIPMENT	0.		
OTHER FUL.	0.		
PAYLOAD		960.	10.28
FUEL-USABLE		1376.	14.74
GROSS WEIGHT		9335.	

TABLE 21. LIFE-CYCLE COST SUMMARY
MUT AIRCRAFT RESIZED TO SAME PAYLOAD

ITEM	DOLLARS	
DEVELOPMENT COST PER AIRCRAFT		88078.
PROTOTYPE COST PER PRODUCTION AIRCRAFT		21981.
RECURRING PRODUCTION COST	520992.	
GFE AVIONICS	40000.	
ENGINE COST	88383.	
(FLYAWAY COST)	(649375.)	
INITIAL SPARES	203373.	
GROUND SUPPORT EQUIPMENT	38962.	
INITIAL TRAINING AND TRAVEL	52496.	
ACQUISITION COST		944206.
FLIGHT CREW	457200.	
FUEL + OIL	295453.	
REPLENISHMENT SPARES	878892.	
ORG + D/S + G/S MAINT	365023.	
DEPOT MAINTENANCE	317258.	
RECURRING TRAINING	273899.	
MAINTENANCE OF GSE	20468.	
OPERATING COST		2608194.
LIFE-CYCLE COST		3662459.
PRODUCTIVITY		.01109
FLEET LIFE CYCLE COST		1831229328.

TABLE 22 . DATA SHEET
MUT AIRCRAFT RESIZED TO SAME PAYLOAD WITH
FREE PLANET TRANSMISSION WEIGHT AND COST

DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G.W. (LB)	9471.	RADIUS (FT)	20.50	RADIUS (FT)	4.40
PAYLOAD (LB)	1027.	CHORD (FT)	1.322	CHORD (FT)	.535
WEIGHT EMPTY (LB)	6552.	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (LB)	1389.	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1178.	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	700.0
HOVER + CLIMB HP	1261.	ASPECT RATIO	15.511	ASPECT RATIO	8.231
MAIN ROTOR DESIGN HP	1048.	CT/SIGMA	.0850	CT/SIGMA	.1089
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9239.1	TAIL ROTOR LIFT	231.9
M.R. DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7147
MAIN G.B. DESIGN HP	1563.	BLADE AREA (SQ.FT)	108.4	BLADE AREA (SQ.FT)	9.4

TABLE 23. SUMMARY WEIGHT STATEMENT MUT AIRCRAFT RESIZED TO SAME GROSS WEIGHT			
GROUP	WEIGHT		% GW
MAIN ROTOR GROUP		820.	8.65
WING GROUP		0.	.00
TAIL GROUP		152.	1.60
TAIL ROTOR/FAN	47.		.49
TAIL SURFACES	105.		1.11
BODY GROUP		1055.	11.14
ALIGHTING GEAR		380.	4.01
FLIGHT CONTROLS		638.	6.74
ENGINE SECTION		100.	1.06
PROPULSION GROUP		1841.	19.44
ENGINES	422.		4.46
AIR INDUCTION	40.		.42
EXHAUST SYSTEM	297.		3.13
LUBRICATING SYSTEM	0.		.00
FUEL SYSTEM	269.		2.84
ENGINE CONTROLS	25.		.26
STARTING SYSTEM	19.		.20
AUXILIARY PROPULSION PROPELLERS	0.		.00
DRIVE SYSTEM	769.		8.12
AUXILIARY POWER UNIT		0.	.00
INSTRUMENTS		135.	1.43
HYDRAULICS		0.	.00
ELECTRICAL GROUP		247.	2.61
AVIONICS		460.	4.86
ARMAMENT GROUP		53.	.56
FURNISHINGS		422.	4.46
AIR CONDITIONING AND ANTI-ICE		48.	.51
AUXILIARY GEAR		60.	.63
VIBRATION SUPPRESSION		76.	.80
TECHNOLOGY SAVINGS		0.	.00
CONTINGENCY		66.	.69
WEIGHT EMPTY		6552.	69.18
FIXED USEFUL LOAD		504.	5.32
PILOT	235.		
COPILOT	235.		
OIL-ENGINE	14.		
-TRAPPED	6.		
FUEL TRAPPED	14.		
MISSION EQUIPMENT	0.		
OTHER FUL.	0.		
PAYLOAD		1027.	10.84
FUEL-USABLE		1389.	14.66
GROSS WEIGHT		9471.	

TABLE 24. LIFE-CYCLE COST SUMMARY
MUT AIRCRAFT RESIZED TO SAME GROSS WEIGHT

ITEM	DOLLARS	
DEVELOPMENT COST PER AIRCRAFT		88483.
PROTOTYPE COST PER PRODUCTION AIRCRAFT		22170.
RECURRING PRODUCTION COST	525581.	
GFE AVIONICS	40000.	
ENGINE COST	89376.	
(FLYAWAY COST)	(654957.)	
INITIAL SPARES	205322.	
GROUND SUPPORT EQUIPMENT	39297.	
INITIAL TRAINING AND TRAVEL	52545.	
ACQUISITION COST		952121.
FLIGHT CREW	457200.	
FUEL + OIL	298318.	
REPLENISHMENT SPARES	885527.	
ORG + D/S + G/S MAINT	367091.	
DEPOT MAINTENANCE	319787.	
RECURRING TRAINING	274179.	
MAINTENANCE OF GSE	20606.	
OPERATING COST		2622708.
LIFE-CYCLE COST		3685481.
PRODUCTIVITY		.01175
FLEET LIFE CYCLE COST		1842740608.

CONCLUSIONS

The free planet transmission concept offers significant improvements over contemporary helicopter transmissions.

The most promising application, from an aircraft point of view, is a drive system for a single-engine, low-horsepower (500 - 600 HP), high-speed engine input. This permits a relatively high reduction ratio free planet unit (14 to 20:1 reduction ratio) and leads to significant reduction in the number of parts in the drive system. If free planet drive is considered for a UTTAS type engine drive train configuration, little improvement can be expected. A utility transport aircraft does not lend itself to design of a high enough ratio free planet drive unit. Engine location requires at least two gear meshes before the free planet unit is reached.

A 9% weight reduction is achieved with the free planet design. The free planet main transmission weighs 671 pounds, compared with 737 pounds for a two-stage conventional planetary design.

The costs of a free planet transmission and conventional planetary transmission are comparable in low quantities. A cost saving of 16.5% can be achieved for a free planet for a production quantity of 500 units.

An improvement in efficiency of over one-half of 1% is achieved through the use of the free planet transmission. Overall main gearbox efficiency is 97.3% for the free planet design and 96.8% for the conventional two-stage planetary. This difference in efficiency translates to a power available difference of 7.5 HP when transmitting 1450 HP to the main rotor.

An improvement in reliability of almost two-to-one is achieved through the use of the free planet design. The predicted MTBF of the free planet unit is 38,600 hours. The conventional two-stage planetary unit has a predicted MTBF of 22,000 hours.

The free planet is more tolerant of loss of lubrication than a conventional planetary design. Improvement is needed in the free planet bearing rings to reduce sensitivity to 23mm AP threats.

Further testing is needed to verify the self-alignment hypothesis and load-sharing characteristics under dynamic conditions.

RECOMMENDATIONS

Further work in the free planet concept for production helicopters should be conducted in an application in which single-engine, high-speed input is available in the 20,000 to 30,000 rpm range while transmitting 400 to 500 HP. This would permit a high-ratio, simple, lightweight transmission.

The self-alignment hypothesis and load-sharing characteristics under dynamic conditions should be verified by strain gaging the planet pinion shafts that orbit the sun gear and rotate about their own centers. Changing the planet pinion indexing tolerances will permit assessment of the actual pinion indexing requirements and may permit significant reduction in unit cost of a free planet transmission.

The effect of loss of normal lubricant supply should be assessed experimentally to verify the expected improvement through the elimination of conventional planetary bearings.

Reliability testing of a free planet unit and conventional two-stage planetary should be conducted to verify the projected 2-to-1 improvement in reliability of the free planet design.

LITERATURE CITED

1. Lastine, James L., ADVANCED TECHNOLOGY VTOL DRIVE TRAIN CONFIGURATION STUDY, Sikorsky Aircraft, Division of United Technologies Corporation; USAAVLABS Technical Report 69-69, U.S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, January 1970, AD 867905.

2. DeBruyne, Neil, A., DESIGN AND DEVELOPMENT TESTING OF FREE PLANET TRANSMISSION CONCEPT, Curtiss-Wright Corporation; USAAMRDL TR-74-27, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, April 1974, AD 782857.

3. Rumbarger, J. H., Leonard, L., DERIVATION OF A FATIGUE LIFE MODEL FOR GEARS, Franklin Institute Research Laboratory; USAAMRDL-TR-72-14, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, May 1972, AD 744504.

4. STRUCTURES MANUAL, Sikorsky Aircraft, Stratford, Connecticut, January 1971.

APPENDIX A

FREE PLANET SELECTION COMPUTER PROGRAM LISTING

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C          PLANET PINION WITH THREE SPUR GEARS PER SHAFT, OUTPUT RING
C          GEAR ON CENTRAL FREE PLANET PINION, AND FIXED RING GEAR ON
C          OUTER FREE PLANET  PINION
C                                   * * * * *
C    * * * * * * *                                           * * * * * * * *
C                                   * * * * *
C          PROGRAM WILL DETERMINE NUMBERS OF TEETH,DIAMETRAL PITCH
C          AND PITCH DIAMETERS FOR ALL MEMBERS,GEAR TOOTH FACE WIDTHS
C          AS WELL AS FREE PLANET UNITS OVERALL LENGTH
C                                   * * * * *
C                                       *
C                                 NOMENCLATURE
C
C                 NOPIN                =  NUMBER OF PINIONS
C                 NSI                  =  NUMBER OF TEETH IN SUN GEAR - INITIAL
C                 NS                   =  NUMBER OF TEETH IN SUN GEAR VARIABLE
C                 PITCH1               =  DIAMETRAL PITCH OF FIRST ROW
C                                         SUN-PINION MESH
C                                         DIAMETRAL PITCHES CONSIDERED
C                                         5,6,7,8,10,12,14
C                 DS                   =  DIAMETER OF SUN
C                 DSMIN                =  DIAMETER OF SUN MINIMUM
C                 NP1MAX               =  MAXIMUM NUMBER OF TEETH IN FIRST
C                                         ROW PINION
C                 NP1                  =  NUMBER OF TEETH IN FIRST ROW PINION
C                 DP1                  =  DIAMETER OF FIRST ROW PINION
C                 NR3MIN               =  NUMBER OF TEETH ON STATIONARY
C                                         REACTION RING GEAR - INITIAL
C                 NR3MAX               =  NUMBER OF TEETH ON STATIONARY
C                                         REACTION RING GEAR - MAXIMUM
C                 NR3                  =  NUMBER OF TEETH ON STATIONARY
C                                         REACTION RING GEAR
C                 NP3MAX               =  NUMBER OF TEETH ON PINION THAT
C                                         MATES STATIONARY REACTION RING
C                                         GEAR - MAXIMUM
C                 NP3                  =  NUMBER OF TEETH ON PINION THAT
C                                         MATES STATIONARY REACTION RING
C                                         GEAR
C                 PITCH3               =  DIAMETRAL PITCH OF THIRD ROW STATION
C                                         ARY RING AND PINION MESH
C                 DR3                  =  DIAMETER OF THIRD ROW STATIONARY
C                                         RING GEAR
C                 DP3                  =  DIAMETER OF THIRD ROW PINION
C                 NR2MAX               =  NUMBER OF TEETH ON SECOND ROW RING
C                                         GEAR-MAXIMUM
C                 NR2MIN               =  NUMBER OF TEETH ON SECOND ROW RING
C                                         GEAR - MINIMUM
C                 NR2                  =  NUMBER OF TEETH ON SECOND ROW
C                                         RING  GEAR
C                 NP2MAX               =  NUMBER OF TEETH ON SECOND ROW
C                                         PINION - MAXIMUM
C                 NP2                  =  NUMBER OF TEETH ON SECOND ROW PINION
C                 PITCH2               =  DIAMETRAL PITCH OF SECOND ROW MESH
C                 DR2                  =  DIAMETER OF SECOND ROW OUTPUT RING
C                                         GEAR
C                 DP2                  =  DIAMETER OF SECOND ROW PINION
C                 RR                   =  REDUCTION RATIO
C                 N                    =  SPEED (INPUT)
C                 NR                   =  SPEED (ROTOR)
C                 FR3P3                =  FACE WIDTH BASED ON COMPRESSIVE
C                                         STRESS OF REACTION RING PINION MESH
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C                    FR2P2                    =  FACE WIDTH BASED ON COMPRESSIVE
C                                                STRESS OF OUTPUT RING PINION MESH
C                    FSUNP1                   =  FACE WIDTH BASED ON COMPRESSIVE
C                                                STRESS OF SUN PINION MESH
C                    AXIAL                    =  AXIAL LENGTH OR HEIGHT OF FREE
C                                                PLANET MEASURED FROM PINION
C                                                EXTREMITY
      REAL   NAPIN
      INTEGER        COUNT
    2 FORMAT   (3F10.0)
    5 FORMAT (1H ,1X,'NS',2X,'NP1',1X,'NP2',1X,'NR2',1X,'NP3',1X,
     1        'NR3',4X,'DS',4X,'DP1',4X,'DP2',4X,'DR2',4X,'DP3',4X,'DR3',
     2         6X,
     2'RR',6X,'N',4X, ,'SP1',1X,'R3P3',1X,'R2P2',1X,'LENGTH',1X,
     3         '1 ROW',2X,'2 ROW',2X,'3 ROW')
    4 FORMAT (1H1,3X,'NUMBERS OF GEAR TEETH',6X,5X,'DIAMETERS OF GEAR',
     1'TEETH   MEMBERS',3X,'REDUCTION',2X,'PIN',1X, 'GEAR FACE'
     2'WIDTH',1X,'AXIAL',2X,'DIAMETRAL PITCH')
      READ (5,2)  DSMIN, HP,RPMOUT
      WRITE (6,2) DSMIN,HP,RPMOUT
      COUNT         = 1
      DO   10 NOPIN = 6,12
      WRITE (6,4)
      WRITE (6,5)
                                                   COUNT      = COUNT + 1
          NSI       = (18/ NOPIN) * NOPIN
      DO   20 NS    = NSI,200,NOPIN
                                                   COUNT      = COUNT + 1
      DO   30     J = 5,12
      PITCH1        = J
                                                   COUNT      = COUNT + 1
          IF (J       .EQ. 11)          GO TO 30
          IF (J       .EQ. 9 )          GO TO 30
          DS        = NS / PITCH1
          IF (DS      .LT. DSMIN )      GO TO 20
          NP1MAX    = 3*NS
      DO   40 NP1   = 18,NP1MAX
C         THE FOLLOWING CARDS CHECK THAT ONLY COMPOUND PLANETARIES
C         ARE CHOSEN THAT HAVE HUNTING TEETH WHICH MEANS FOR EACH
C         MESH WE MUST SHOW THAT
C
C                       NS/NP = WHOLE NUMBER +UNREDUCIBLE FRACTION
C                       NR/NP = WHOLE NUMBER +UNREDUCIBLE FRACTION
C                 ****                                 ****
C         CHECK FOR HUNTING TEETH OF P1-S1 MESH
C                 ****                                 ****
          L1        = NS
          N1        = NP1
    3     K1        = L1/N1
          K4        = L1-K1*N1
          IF   (K4 -1) 1,6,7
    1     K2        = 0
          GO TO       9
    7     L1        = N1
          N1        = K4
          GO TO       3
    6     K2        = 1
    9     CONTINUE
          IF   (K2 .EQ. 0)   GO TO 40
                                                   COUNT      = COUNT +1
          DP1       = NP1 / PITCH1
      NAPIN         = NOPIN
          IF(( 180./ ASIN  (( DP1 + 2./ PITCH1 )/(DS + DP1))).LT.
     1    NAPIN)                         GO TO 30
          NR3MIN    = (NS + 2* NP1) / 2
          IF (NR3MIN   .LT. 60 )   NR3MIN   = 60
          NR3MIN    = (NR3MIN / NOPIN ) * NOPIN
          NR3MAX    = 5 *NR3MIN
```

```
           IF (NR3MAX  .GT.200 )  NR3MAX  =200
      DO  50 NR3    = NR3MIN,NR3MAX,NOPIN
      NP3MAX        =(NR3*2)/5
                                                        COUNT      = COUNT +1
      DO  60 NP3    = 18,NP3MAX
                                                        COUNT      = COUNT +1
C             ****                                     ****
C          CHECK FOR HUNTING TEETH OF R3-P3 MESH
C             ****                                     ****
           L1       = NR3
           N1       = NP3
  403      K1       = L1/N1
           K4       = L1 - K1*N1
           IF   (K4- 1) 401,404,402
  401      K2       = 0
           GO TO    405
  402      L1       = N1
           N1       = K4
           GO TO    403
  404      K2       = 1
  405      CONTINUE
           IF   (K2 .EQ. 0)   GO TO  60
           PITCH3   = (NR3 - NP3)/(DS + DP1)
           IF  (PITCH3 .GT.12.0) GO TO 60
           [illegible]
    1      K2       = 0
           GO TO       9
    7      L1       = N1
           N1       = K4
           GO TO       3
    6      K2       = 1
    9      CONTINUE
           IF   (K2 .EQ. 0)   GO TO 40
                                                        COUNT      = COUNT +1
           DP1      = NP1 / PITCH1
     NAPIN          = NOPIN
           IF(( 180./ ASIN   (( DP1 + 2./ PITCH1 )/(DS + DP1))).LT.
    1      NAPIN)                            GO TO 30
           NR3MIN   = (NS + 2* NP1) / 2
           IF (NR3MIN  .LT. 60 )  NR3MIN   = 60
           NR3MIN   = (NR3MIN / NOPIN ) * NOPIN
           NR3MAX   = 5 *NR3MIN
           IF (NR3MAX  .GT.200 )  NR3MAX  =200
      DO  50 NR3    = NR3MIN,NR3MAX,NOPIN
      NP3MAX        =(NR3*2)/5
                                                        COUNT      = COUNT +1
      DO  60 NP3    = 18,NP3MAX
                                                        COUNT      = COUNT +1
C             ****                                     ****
C          CHECK FOR HUNTING TEETH OF R3-F3 MESH
C             ****                                     ****
           L1       = NR3
           N1       = NP3
  403      K1       = L1/N1
           K4       = L1 - K1*N1
           IF   (K4- 1) 401,404,402
  401      K2       = 0
           GO TO    405
  402      L1       = N1
           N1       = K4
           GO TO    403
  404      K2       = 1
  405      CONTINUE
           IF   (K2 .EQ. 0)   GO TO  60
           PITCH3   = (NR3 - NP3)/(DS + DP1)
           IF (PITCH3 .GT.12.0) GO TO 60
           IPITCH3  = PITCH3
```

```
            RPICH3   = IPICH3
            IF   ( PITCH3 .GT.(RPICH3+.001))    GO TO 60
            IF   ( PITCH3 .LT.(RPICH3-.001))    GO TO 60
            DR3       =  NR3 / PITCH3
            DP3       =  NP3 / PITCH3
            IF   ((DR3 -(2.* DP3)) .LT.     (DSMIN-1.0))  GO TO 60
            IF  (DR3 .GT. 13.0)            GO TO  60
       NAPIN         = NOPIN
            IF (( 180./ ASIN  ((DP3 +2./PITCH3) /(DR3 - DP3))) .LT.
     .      NAPIN)                       GO TO 50
            NR2MAX   = 2 * NR3
            NR2MIN   = NR3 / 2
            IF   ( NR2MIN .LT. 60 )    NP2MIN = 60
            NR2MIN   = (NR2MIN / NOPIN)* NOPIN
       DO   70 NR2    = NR2MIN,NR2MAX,NOPIN
                                                        COUNT      = COUNT +1
            NP2MAX   =(NR2*2)/5
       DO   80 NP2    = 18,NP2MAX
                                                        COUNT      = COUNT +1
C               ****                                   ****
C           CHECK FOR HUNTING TEETH OF R2-P2 MESH
C               ****                                   ****
            L1        = NR2
            N1         = NP2
  303       K1         = L1/N1
            K4         = L1-K1*N1
            IF     (K4 -1) 301,304,302
  301       K2         =  0
            GO TO    305
  302       L1        = N1
            N1        = K4
            GO TO    303
  304       K2        = 1
  305       CONTINUE
            IF (K2 .EQ. 0)  GO TO 80
            PITCH2   = (NR2 - NP2 )/(DS + DP1 )
            IF (PITCH2 .GT. 12.0)GO TO 80
            IPICH2   = PITCH2
            RPICH2   = IPICH2
            IF ( PITCH2 .GT.(RPICH2+.001 ))    GO TO 80
            IF ( PITCH2 .LT.(RPICH2-.001 ))    GO TO 80
            DR2       = NR2/ PITCH2
            DP2       = NP2/ PITCH2
            IF   ((DR2 -(2.*DP2)) .LT.      (DSMIN- 1.0))   GO TO   80
            IF   (DR2 .GT. 19.0)           GO TO 80
       NAPIN         = NOPIN
            IF (( 180./ASIN   ((DP2 + 2./PITCH2) /(DR2- DP2))).LT.
     1      NAPIN)                        GO TO   70
       AS             = NS
       AP2            = NP2
       AR2            = NR2
       AR3            = NR3
       AP1            = NP1
       AP3            = NP3
            IF  ((NR3*NP2).EQ.(NP3*NR2))  GO  TO  201
            RR        = ( 1 + (AR3* AP1)/(AP3* AS))/ (1 - ( AR3 *AP2) /
     1                 ( AP3 * AR2))
            GO TO 202
  201       RR        =  0.0
  202       CONTINUE
       IF   ( RR .LE. 0.0)              GO TO 80
            IF  (( ABS(RR).LT.10.).OR.      (ABS(RR).GT. 30.0))  GO TO 80
            RPMIN    = RR* RPMOUT
            TORQS    =(63025.*HP)/RPMIN
            TORQR2   =(63025.*HP)/RPMOUT
            WTSUN    = (2.0*TORQS)/(DS*    NAPIN)
            WTR2     = (2.0*TORQR2)/(DR2*  NAPIN)
```

```
          WTR3      = WTR2 -WTSUN
          FSUNPI    = ((21.*WTSUN)*       (1./DS+1./DP1))/ ((SIN(2.*22.5))
     1*(  145.**2))
          FR3P3     = ((21.*WTR3)*(1./DP3-1./DR3))/ ((SIN(2.*22.5))*(145.
     1 **2))
          FR2P2     = ((21.*WTR2)*(1./DP2-1./DR2)) / (( SIN(2.*22.5))*(14
     15.**2))
          IF   ( FR2P2 .GT. (.50*DP2))  GO TO  80
          IF (( DP2.GT.DP3).AND.( DP1 .LT. DP2 ) )  GO TO 101
          T         =  FR3P3 /2.0 *.1D * FR2P2 / 2.
          A         = (DP2*T )/(DP3-DP2)
          B         = (DP1/DP3)*(A*T)
          IF((A*B).LT.( T-.1))      GO TO 101
          AXIAL     = T * A * B * FR3P3/2.* FSUNPI/2.
                                               GO TO 102
  101     AXIAL     = 0.0
  102     CONTINUE
          IF ( AXIAL .GT. 13.0) GO TO 80
          IF ( AXIAL .GT. (3.* DR3))   GO TO 80
      WRITE (6,90)    NS,NP1,NP2,NR2,NP3,NR3,DS,DP1,DP2,DR2,DP3,DR3,RR,
     5       NOPIN,   FSUNPI,FR3P3,FR2P2,AXIAL,
     6                PITCH1,PITCH2,PITCH3
   90 FORMAT (1H ,I4,5I4,2X,F5.2,2X,F5.2,2X,F5.2,2X,F5.2,2X,F5.2,2X,F5.2
     1        ,1X,F11.3,1X,I2,2X,F4.1,1X,F4.1,1X,F4.1,1X,F6.2,1X,F6.3,1X,
     2         F6.3,1X,F6.3)
   80 CONTINUE
   70 CONTINUE
   60 CONTINUE
   50 CONTINUE
   40 CONTINUE
   30 CONTINUE
   20 CONTINUE
   10 CONTINUE
      WRITE (6,1000)  COUNT
 1000 FORMAT(1H ,'THE NUMBER OF INTERATIONS IN ARRIVING AT A SOLUTION IS
     1',I20)
      STOP
      END
@XQT
  5.0       1385.0      345.0
@FIN
```


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